FLUID POWER
WITHOUT MOVING PARTS

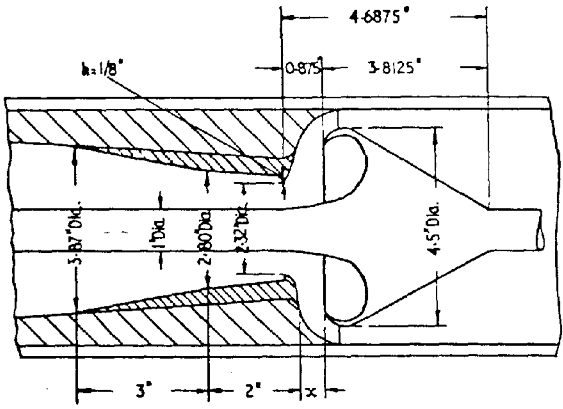

FIG. 15a. MODIFICATION 1
REDUCED THROAT

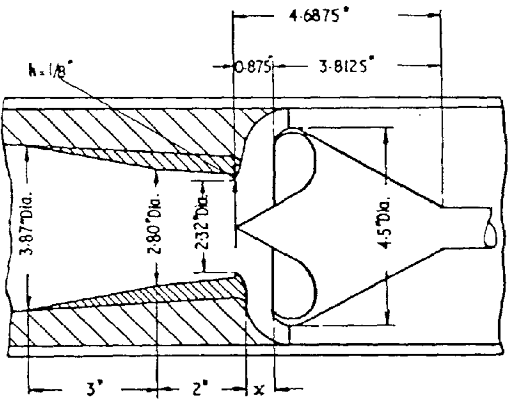

FIG. 15b. MODIFICATION 2
CENTRE BAR REMOVED

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PRINCIPLES OF REFRIGERATION, PART 3

by C.T. Baker and the I.C.S. Staff
c 1937 International Textbook Co.
p. 31-36

STEAM-JET SYSTEM OF REFRIGERATION

40. **Principle of Steam-Jet Refrigeration.**—Another type of refrigeration system that is used is known as the vacuum, or steam-jet, system. Ordinary water forms the refrigerant, and the evaporation of the refrigerant is caused by lowering the pressure on it, and consequently reducing the temperature at which it will vaporize. This reduction of pressure is produced by the action of steam jets of high velocity.

A diagram of the system, showing the several parts and illustrating the method of operation, is given in Fig. 11. The water to be chilled flows into the air-tight vessel *a* through the pipe *b* and is discharged from the header *c* in a fine spray by a series of spray heads. A very high vacuum is produced in the vessel *a* by the action of a steam-jet ejector *d* that draws out the air and vapor in the vessel above the level *e* of the water. In other words, the pressure inside the vessel is reduced to only a fraction of a pound, absolute; and as this pressure is lower than that at which the water will boil, some of the water at once flashes into vapor. The heat required to cause this evaporation is taken from the water in the vessel, and thus the temperature of that water is reduced.

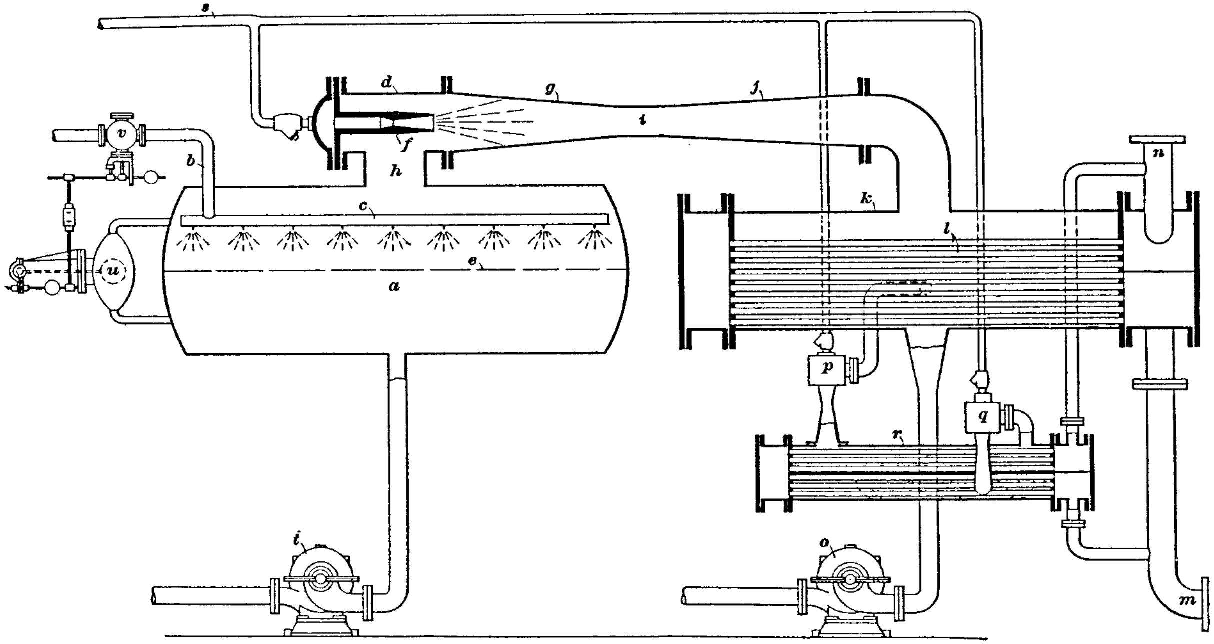


FIG. 11

41. The amount of cooling possible may be very easily shown by an example. Suppose that the water enters the vessel *a*, Fig. 11, at a temperature of 80° F., and that a pressure of .4 pound, absolute, corresponding to a vacuum of about 29 inches of mercury, is maintained in the vessel. At that pressure, water will boil at a temperature of 72.86° F., say 73° F. Consequently, when the water at 80° F. enters the vessel, about $3\frac{1}{2}$ per cent. of it will flash into steam; that is, from every pound of water entering, about .035 pound is vaporized. The latent heat of vaporization of water at a pressure of .4 pound is very nearly 1,053 B. t. u. per pound, and so the vaporization of .035 pound will absorb $.035 \times 1{,}051 = 36.9$ B. t. u. This 36.9 B. t. u. removed from the water reduces the temperature of each pound from 80° F., evaporates about .035 pound at 73° F., and chills the remaining .965 pound. Reduction of the temperature of 1 pound through $80 - 73 = 7$ degrees takes 7 B. t. u., and so the amount left for chilling is $36.9 - 7 = 29.9$ B. t. u. Hence, the .965 pound of water remaining is lowered $29.9 \div .965 = 31$ degrees, nearly, which means that the water in the vessel is chilled to a temperature of $73 - 31 = 42°$ F.

42. The operation of the ejector *d*, Fig. 11, known as the booster ejector, in removing air and vapor from the upper part of the vessel *a* is based on the entraining action of the jet of steam escaping at high velocity from the nozzle *f*. Steam expands while passing through the nozzle, thereby changing its pressure to velocity. The high velocity steam picks up the flashed vapor from the vessel *a* and carries it from the space *h* around the nozzle to the converging section *g* of the ejector, the mixture of air and vapor being discharged through the narrow throat *i* into the diverging pipe *j*. In so doing, the velocity of the mixture is converted into pressure. The ejector is so designed that the pressure attained in this manner is sufficiently high to enable the vapor to be condensed in the condenser *k* by the available condensing water. The vapor fills the interior of the cylindrical condenser shell, surrounding the tubes *l* through which the condensing water flows. This water enters through the pipe *m*, makes two passes through the condenser, and escapes at *n*, carrying with it the heat of the discharged steam. The steam is thus condensed, and the great decrease in volume following condensation assists in producing and maintaining the vacuum in the condenser and in the vessel *a*. The water of condensation, or condensate, is removed by the pump *o*.

43. Because of the fact that the vacuum obtained must be high, secondary ejectors p and q, Fig. 11, are added, their purpose being to insure the removal of the last traces of air from the evaporator. The ejector p is smaller than that at d and entrains air from the condenser k, discharging its mixture of steam and air into the upper compartment of a second condenser r, where the steam is condensed. The smallest ejector q draws out the air from the upper compartment of the condenser r and discharges it into the lower and separate compartment of the condenser r, from which the air escapes to. the atmosphere together with the small amount of water from the condensation of the steam used to operate the ejector q. The effect of these three ejectors in series, together with the condensers, is to produce and maintain in the evaporator a vacuum closely approaching a perfect vacuum. All three steam-jet ejectors receive steam from the supply pipe s, and the condenser r receives its water supply from the pipe m and discharges into the pipe n.

44. The water that has been chilled in the evaporator a, Fig. 11, flows to the pump t, by which it is forced to the point where the desired cooling is to be done. The level e of the water in the evaporator is maintained by an automatic regulator. A float u in a float chamber connected to the evaporator falls or rises as the water level changes, and through a system of connecting links causes the valve v on the water supply pipe to open or close. Thus a fall of the level increases the inflow and a rise decreases it, so that the level is kept fairly constant.

45. It will be observed that there are no moving parts in the refrigerating system shown in Fig. 11. Moreover, the refrigerant is harmless in every way and there are no dangerous pressures. The first cost of the apparatus and the cost of operation compare favorably with those for mechanical refrigeration of the compression type. It is not necessary to provide steam at high pressure. It is possible to produce a freezing temperature in the evaporator, but this can be done only at the expense of using large quantities of steam and water. The cooling water for the condensers may be taken from the city main, a well, or a cooling tower.

46. In a modification of the system just described, the vapor after being abstracted from the evaporator is compressed before it enters the condenser. Units of this type are more and more being used for air-conditioning purposes and are built with a refrigerating capacity ranging from less than 1 ton to 150 tons. The operation of a system of this kind is similar to that of any other compression refrigerating system. After water has been vaporized in an evaporator, the vapor is compressed and then condensed at a higher pressure, the vacuum produced in the condenser by the circulating water being less than that existing in the evaporator. In a typical example of such a system the water is vaporized at 50° F. at an absolute suction pressure of .178 pound per square inch. Then, the vapor is compressed to approximately .82 pound per square inch, absolute and condensed at a circulating-water temperature of about 90° F.

Because of the very low absolute pressures employed in this system care must be taken to accomplish efficient removal of the vapor from the evaporator. When steam at a pressure of 50 pounds or higher is available, this may be effected by the use of a steam-jet ejector. In other cases, when the available steam pressure is less than 50 pounds, a small rotary vacuum pump direct-connected to an electric motor is used instead of the steam-jet ejector. The compressor used may be of either the reciprocating or the rotary type and may be driven by a direct-connected electric motor or condensing steam turbine. In the latter case the turbine condenser generally uses circulating water discharged from the vapor condenser, thereby effecting a material saving in cooling water.

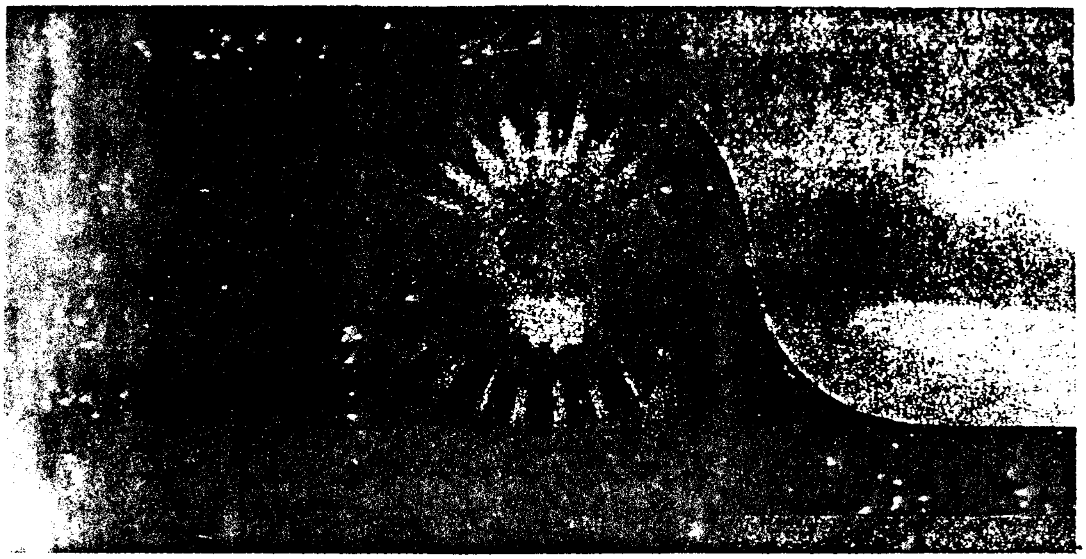
Basic Refrigeration, by Guy R. King
c 1951 Nickerson & Collins Co., p. 43

REFRIGERANTS

Water.—To compare the list of requirements of a good refrigerant with the properties of water would seem to make a very poor refrigerant of this common fluid. The enormous volume of its vapor at required evaporator temperatures is such that the use of an ordinary compressor is impossible. High vacuums are required in both the low and high pressure sides of the system. The required vacuum on the low side is so high that it is difficult to maintain. Leaks are difficult to find. Their only evidence is when air leaking into the system prevents proper operation. To check for a suspected leak the system must be charged with compressed air and all joints tried wth soap suds. The 32 F freezing point makes it impossible to use it for ordinary applications.

However, for certain applications above 32 F such as chilling large quantities of water and for air conditioning, it has satisfactorily filled some very rigid requirements. First, it is absolutely safe. It has no odor or poisonous properties, and it is not inflammable. If any amount of it were accidentally released in a crowded building, it would not cause panic. It is cheap and can be obtained anywhere.

The problem of handling the excessive volume of the low pressure vapor has been solved by the development of centrifugal compressors and steam ejectors, either of which will easily handle large volumes of vapor at low pressure. The steam jet system using water as a refrigerant has been used extensively in the air conditioning of passenger trains. Its lack of hazard and availability, plus the absence of mechanical compression equipment, makes it a very desirable refrigerant where the rough service of railroad conditions makes leaks more probably than in stationary systems.



HEAT PIPES, such as above from Battelle Memorial Institute, are used to move solar heat quickly, efficiently and passively.

A Self-Powered Solar Heat Pipe

By Charles A. Miller

WE KNOW a lot about collecting and storing solar energy. However, a lingering problem and one that is getting new attention these days is how to transport it efficiently from place of collection to place of use.

One gadget that promises to make solar doings more efficient is called a *heat pipe*, a little conduit that, handily enough, takes heat from where it's collected and moves it to where it's used. And does it better than conventional pumping systems. Best of all, the heat pipe is entirely self-powered.

The heat pipe idea was patented way back in 1944 by Frigidaire but the whole thing was largely ignored for the next three decades. Heat pipes did gradually find small jobs in industry and aerospace but most people never heard of them.

One of those small jobs turned out to be on the Alaska Pipeline—and it wasn't exactly small. Along much of the line's run, finned tubes jut out atop the pipe supports like reindeer antlers. They are heat pipes out there to siphon heat away from the permanently-frozen tundra so it stays that way rather than melting and becoming a bog. Pictures showing the antler-like heat pipes appeared by the millions in the press of the world but, so far as we know, never got identified as such. Nobody cared. So the heat pipe labored on, a much-seen but anonymous hero of the North.

[illegible] transferred to interior of [illegible] pipes. Fluid returns by [illegible] pushed along by grooves.

Things seem to be changing fast now, however, because it has finally dawned on researchers that the heat pipe could be a dandy device for transporting solar energy. That may be like the touch of Midas for the lowly heat pipe. Solar devices these days are getting movie-star treatment. Specifically, the Battelle Memorial Institute, a research organization in Columbus, Ohio, is beginning to experiment with what is being called the solar heat pipe.

What is a heat pipe? Basically, it's a device of transport, just as is a gas pipe. In general looks, it's not much different, either, being nothing but a hollow tube. With one exception: It's sealed at both ends. A heat pipe could be the size of a ballpoint pen or as large as a hotrodder's tailpipe. Usually they're of small bore (less than an inch) and no more than a few feet long, depending on how they're used. The practical limit on length is about 60 ft.

The one unique attribute of a heat pipe is inside its walls. Sealed in there is a small amount of fluid that may be water, ammonia, acetone or some other liquid.

The way a heat pipe works is simple. When one end is heated, the liquid inside boils, turning to vapor. The vapor travels to the other end of the pipe where it condenses, giving off heat in the process. The condensate then returns to the end where it started to repeat the cycle.

Heat is transferred over the length of the pipe via the moving vapor.

Because only a small amount of liquid is needed to sustain the cycle, the process starts quickly once the pipe is exposed to heat.

How well does a heat pipe do its thing? We got hold of an experimental heat pipe from Battelle Memorial Institute to see for ourselves.

Our pipe is representative of most heat pipes. It's sealed at both ends, the air inside is low pressure and the working fluid is water. The tube is grooved inside to help move the water back to its origin after it's condensed.

To try it out, we stuck one end of the pipe in a cup of hot water. Seconds later, we could hear the water in there boiling. And almost as quickly, the other end of the pipe was too hot to handle. It worked like a charm.

As we said earlier, the fluid inside can be most any liquid, but certain fluids are better than others. Water, for instance, is a good medium and can be made to have a lower boiling point to suit the lower temperatures of flat-plate collectors, the most common type solar collector. How?

One of the basic principles of physics that applies to heat pipes using H_2O is the fact that when you lower the pressure on a liquid, the boiling point goes down, too. For example, water boils at a lower temperature at the top of a mountain than it does at the bottom. If you cook potatoes at your cabin in the hills, you'll notice it takes very little time to get the water boiling, compared to your home in the valley.

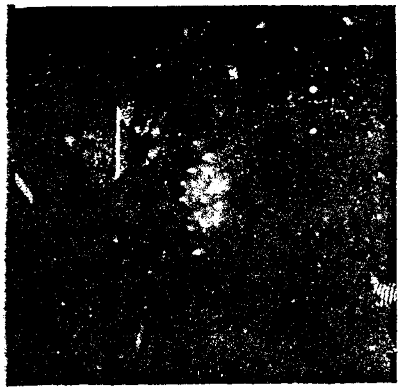
The same idea applies to a heat pipe. When the pressure is lowered inside the sealed tube, the liquid boils at a lower temperature, which is ideal over the range of temperatures used to heat homes with solar energy—from 100°. to 200°F, normally, with flat-plate collectors.

Of course, other types of fluids, especially those that don't freeze, may be better suited for flat-plate collectors.

The basic principle of heat pipes is simple, but the simplicity is almost deceiving. There are a few difficulties that sometimes muck up the works.

For instance, the Liquid must be pure. If there are any impurities in it, the working range of temperatures is affected. And getting the liquid back to its starting point can be tricky, unless the pipe is near vertical.

The pipe can be used, theoretically, in any position, but it's most effective when it's almost upright. That's because gravity has a hand in its operation—a helping hand, that is. The main job of getting the liquid back to its starting point is often accomplished by adding grooves to the inside of the tube, or a mesh, or some other material to act as a wick.



EQUIPMENT to test effectiveness of heat pipe at Battelle Memorial Institute. Heat is measured along length of pipe.

How do heat pipes fit into the solar energy scene? Like a hand in a glove.

In a conventional solar system, the heat that is collected from the sun is transported by means of a working fluid or air, depending on the type of system. We're all familiar with the flat-plate liquid collector that sits on the roof. Ordinarily it's filled with a nonfreezing fluid and connected by plumbing to a heat exchanger and pool of water or bin of rocks in the basement to store the sun's energy. (Or the heat of the sun is concentrated by a parabolic collector, then transported to storage.)

A solar collector using a heat pipe as a means of moving heat would be configured differently, however.

One envisioned by Battelle, for example, would have the solar collector panel incorporated into a wall facing south. Connected to its rear side would be a series of heat pipes.

Say the sun, in Battelle's solar setup, is shining brightly, heating the collector to a working temperature. The collector, in turn, is transferring the sun's warmth to the heat pipes. Inside the heat pipe, the working fluid is boiling, turning into a vapor that travels up the length of the pipe, through insulation to the other end, which is embedded in a concrete wall or pool of water that is storing the heat.

All of a sudden, the sun goes behind a cloud. The source of heat is cut off.

Heat would ordinarily begin to radiate from storage back to the now cooler collector, wasting the heat. Same thing would happen at night—all night is a matter of fact. The heat pipes, however, prevent that from happening. Heat can't travel back from the wall to the collector because the working fluid has moved back to the collector. There's no fluid at the top of the pipe to vaporize. So the heat pipe acts as a one-way switch, shutting down when the sun goes away.

One of the beauties of heat pipes is that they can do their thing without the need for a lot of hardware to control them. No switches, photocells, valves and whatnot. And they are reliable.

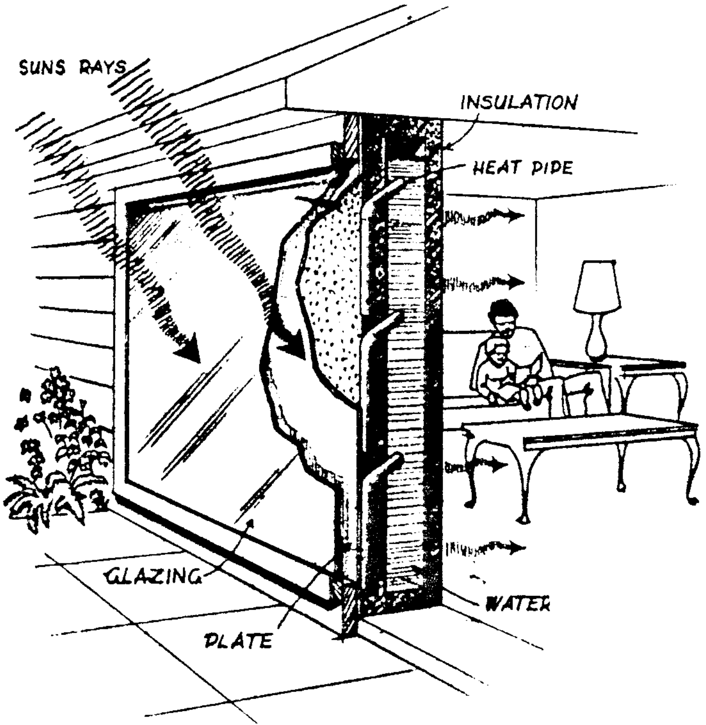
Heat pipes, it could be said, are the 18-wheelers appearing on the solar energy scene. They neither collect nor consume—only transport.

And because they move heat, they can be used for cooling, too, by taking heat away from its source. As a matter of fact, there are some industrial applications of this phenomenon already. Hard-to-reach bearings are sometimes cooled with heat pipes. Someone even once made an oil dipstick out of a heatpipe, using it to cool the oil in a motorcycle crankcase.

Since heat pipes convey heat so well, they have already been put to other types of practical use. They've been used to keep snow from accumulating on highways. Embedded vertically to a depth of 30 to 40 ft. below a roadway, they bring heat up from underground to melt the white stuff.

They have been used, too, in medicine, mostly for cryogenic surgery (cutting by freezing). They are used in space to cool spacecraft components and in industry to equalize temperatures of boiler equipment.

But since they are such a high-performance heat mover, their use in solar applications is a natural. Heat pipes can take advantage of the small temperature differences in passive collecters and move collected heat into storage very quickly. And because they are, in effect, diodes for heat, once the heat is in storage, it stays there.



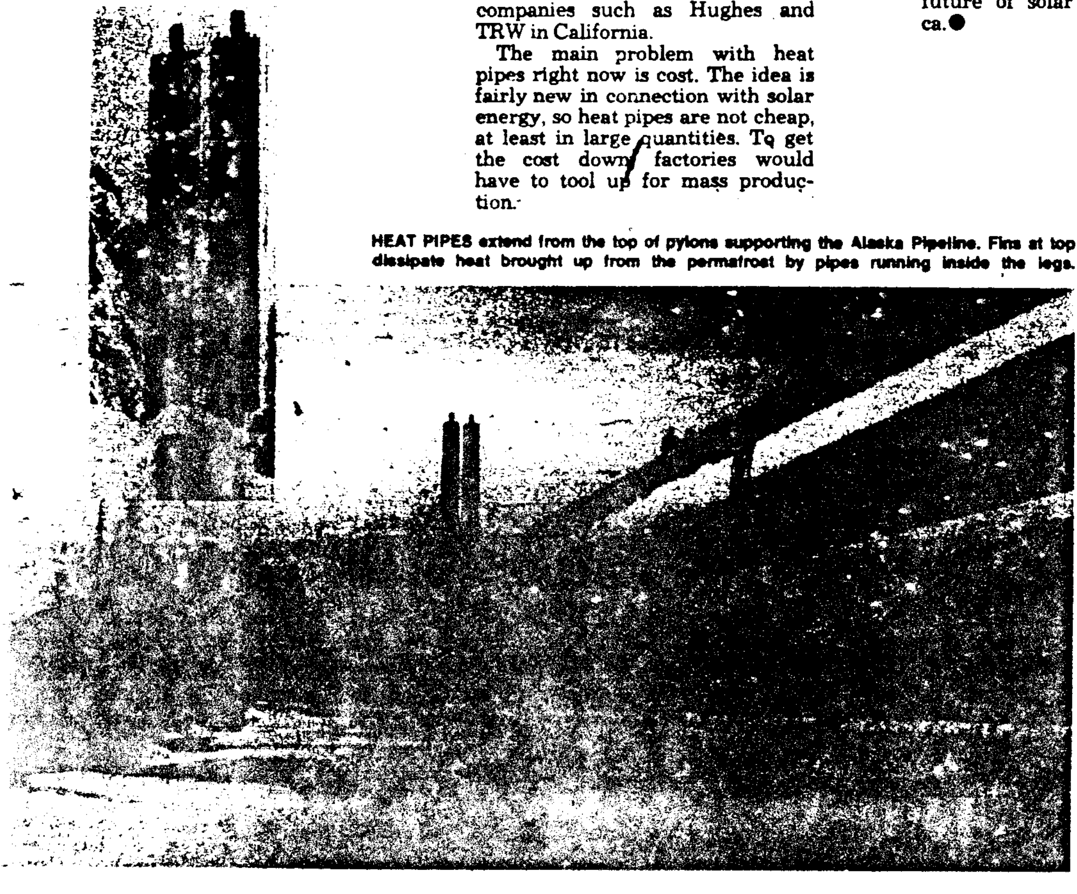
PROPOSED solar system, using heat pipes as the medium for transferring heat from plate to storage. Heat travels in one direction only. The system is entirely passive.

There are a number of organizations working to find the right niche for heat pipes in solar applications. Besides Battelle, Thermacore, Inc., has been looking into the use of heat pipes in evacuated-tube solar collectors made by GE. There also are study programs at Los Alamos Lab, the University of New Mexico and at aerospace companies such as Hughes and TRW in California.

The main problem with heat pipes right now is cost. The idea is fairly new in connection with solar energy, so heat pipes are not cheap, at least in large quantities. To get the cost down factories would have to tool up for mass production.

Because the solar industry is still in its infancy, any new technology that comes along costs plenty to develop. And since there are tried and true methods for moving heat, like conventional plumbing and ducting, the heat pipe is considered an expensive alternative. But that could change as research pays off.

Moving heat from one point to another is not a new idea, but heat pipes do it so well and with so little hardware, their potential for solar applications is tremendous. Heat pipes could be a large part of the future of solar power in America. ●



HEAT PIPES extend from the top of pylons supporting the Alaska Pipeline. Fins at top dissipate heat brought up from the permafrost by pipes running inside the legs.

6

★HEAT PIPES

(see Heat Transfer Equipment)

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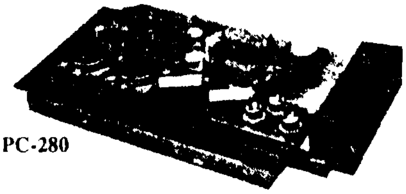

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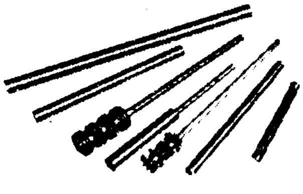

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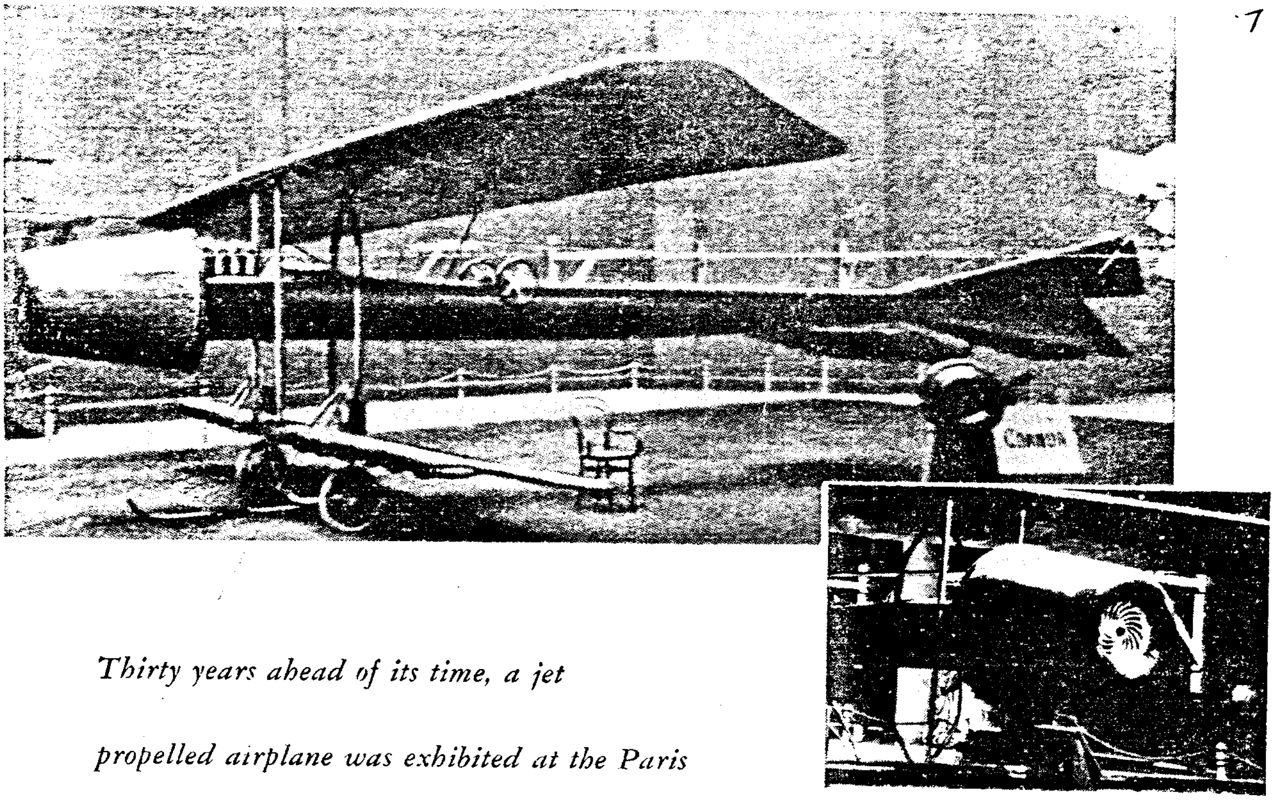


The Coanda biplane featured all-wood covering, remarkably clean lines and jet propulsion. Inset: Controversial turbine engine.

Thirty years ahead of its time, a jet propelled airplane was exhibited at the Paris Salon d'Aeronautique to become the

GRANDADDY OF JETS

By **JOSEPH BONANNO**

THE PARIS AVIATION exhibit of 1910 was a success. Only seven years previously, the Wrights had launched the age of flight, but in that time the French had taken to the new element with vigor. Names appeared on the French horizon to rival Wright and Curtiss, names like Bleriot, Latham, Vedrines, and Farman. Even the genial Brazilian aeronaut, Santos-Dumont, and the Englishman, Farnham, left their own countries to build and fly their machines in France.

Bleriot had flown the English Channel just the year before, symbolizing the strengthening bond between France and England, but suggesting too, the end of Britain's island impregnability. And in September 1910, just a month before the Paris Salon, the Frenchman, Chavez, had flown the alps. Aviation had come a long way since 1903, but nowhere further than in France.

So it was with mingled admiration, envy, and enthusiasm that the air enthusiasts of the world flocked to Paris in October of 1910 for the Salon D'Aeronautique. They went expecting to see the best and were not disappointed. The *Nieuport*, the *Caudron*, the *Goupy*, the *Antoinette*, the *Breguet*, the *REP*, the *Astra* and other flying machines were on display, machines that helped the French set the pace of air progress in the world of 1910. But the crowd saw even more than it expected to see.

For there was in the exhibit a new and strange aircraft, designed by Henri Coanda.

The first thing about Monsieur Coanda's biplane that caught the eye was its remarkably clean lines, the almost complete absence of wires and struts.

The tail assembly looked like no tail assembly before it. In place of the traditional fins, M. Coanda substituted a configuration of four tail-planes set at a 45 degree angle to the fuselage and at right angles to each other, resembling a St. Andrew's cross. Attached to the broad end of each of the four surfaces was a hinged triangular flap. These flaps were controlled by wheels on either side of the cockpit.

The airfoil M. Coanda designed had a conventional curvature of the upper surface but featured two curves of different radii on the underside. Under the curved surface were a number of short projecting ribs which began at the leading edge and tapered to a point before reaching the trailing edge. These, said M. Coanda, insured lateral stability by keeping the airflow under the wings parallel.

The span of the lower wing was considerably shorter than that of the upper, and neither was attached directly to the fuselage. Actually, more than two feet separated each wing from the fuselage. Conventional struts were absent. Two strong steel pipes on either side of the fuselage held the wings together. The entire aircraft was covered with a thin, highly polished wood veneer.

The landing gear was the most conventional section of Monsieur Coanda's airplane, (*Continued on page 44*)

Flying 1-54 didn't get the rest

Henri Coanda's Answer to the Engine:

Who Needs Engines?

R.H. Warring   Pneumatic Handbook, 1982

Airmovers

THE JETFLOW Airmover is a device which converts a small quantity of low pressure air into a large volume of fast moving air, its principal application being auxiliary ventilation of confined areas, or localized ventilation. Since the device works without power (other than a supply of low pressure air) or moving parts, it can provide safe ventilation in potentially explosive atmospheres.

The principle of operation is shown in Fig 1. A supply of primary air enters the manifold where it expands to induce a depression at the throat of the venturi section and also attaches itself to the current aerofoil surface by Coanda effect. Secondary air from the surroundings is induced into the throat of the manifold, where it mixes with the primary air and accelerates with it through the divergent tube. The result is a high velocity mixture of primary and secondary flows emitted from the end of the tube, entraining more surrounding air.

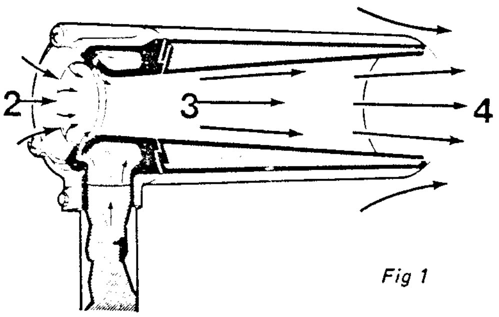
Jetflow Airmovers can be operated singly, in series or in parallel and they perform extremely well when connected to ducting. There is, however, a limit to the resistance they will overcome but at best the Airmover will generate a static pressure (suction) of about 112 m/bar (45" W.G.). The resistance to airflow is determined by the length and diameter of the ducting and the frictional characteristic (K factor) of the duct lining.

Adequate Airmover performance depends on the correct application of these variables:-

(a) Gap Setting.

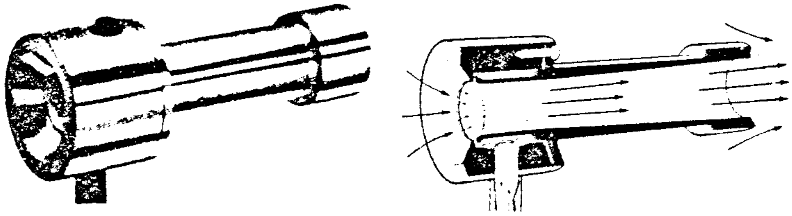
(b) Pressure of primary supply in manifold.

(c) Volume of primary supply available.

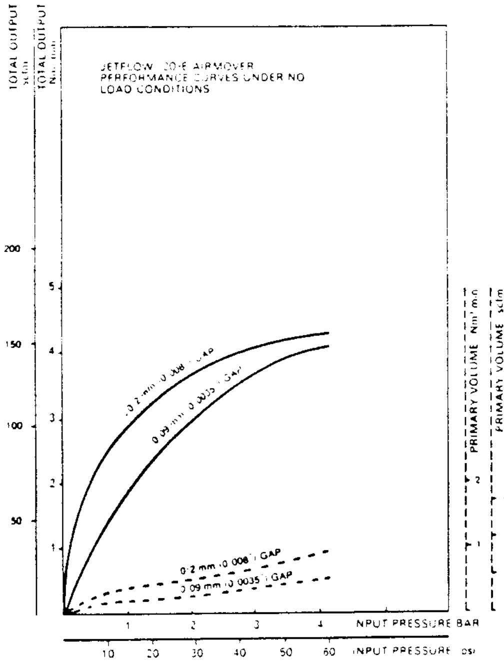


1 — *Primary supply enters the manifold via annular gap and accelerates over the aerofoil.*
2 — *Secondary supply is induced into the throat of the manifold.*
3 — *Mixing, cooling and diluting in the divergent tube.*
4 — *Expelled at high velocity entraining surrounding air.*

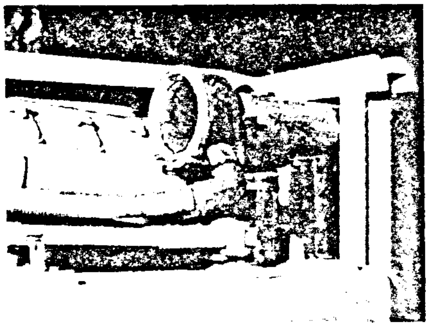
Fig 1



Olin 'Jetflow' 20 air mover.

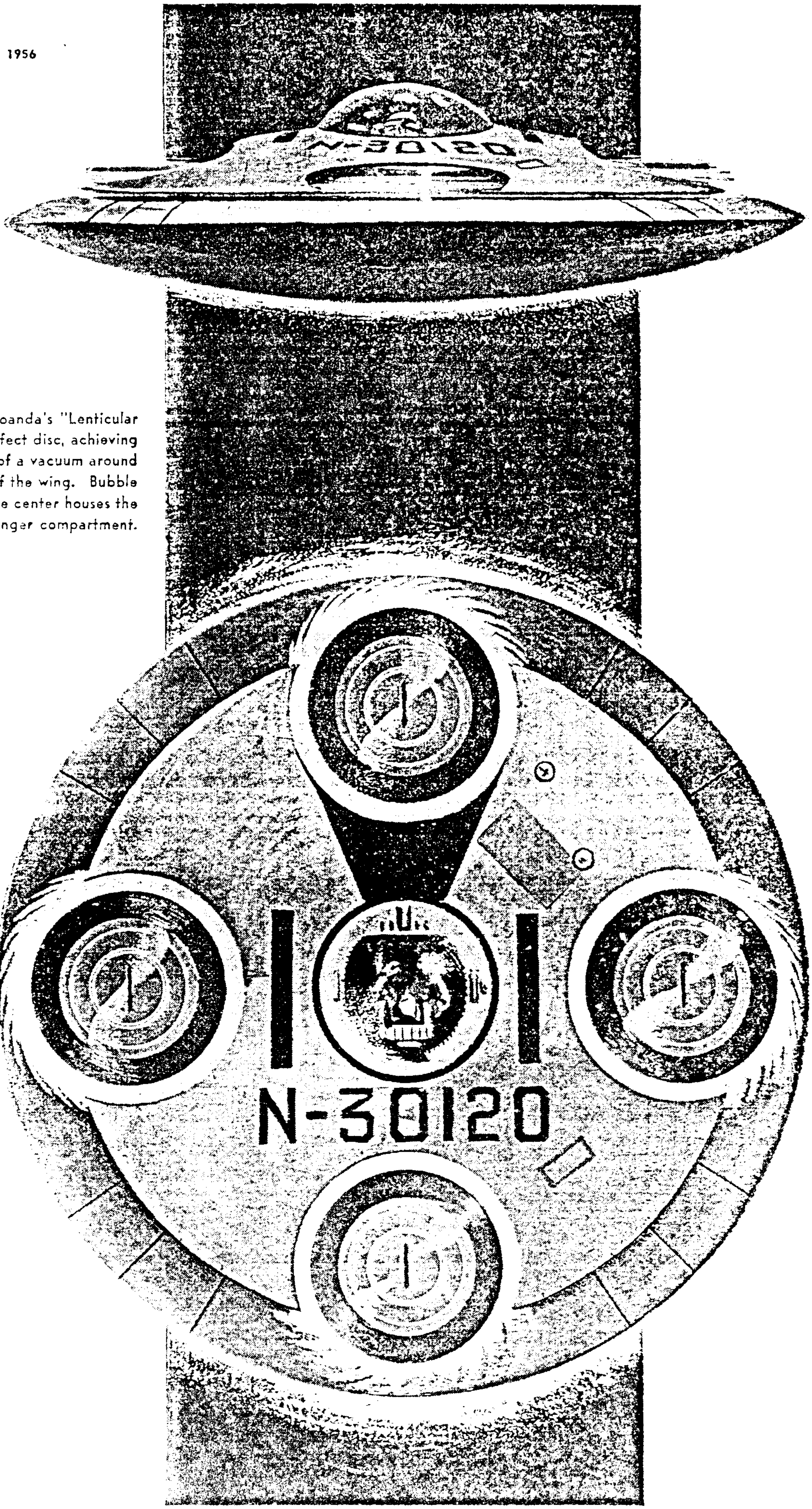


Performance characteristics of Jetflow 20-E Airmover.



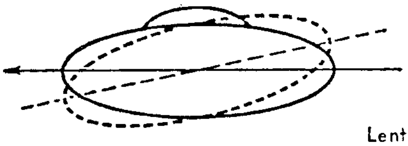
Typical installation of a 'Jetflow Airmover' on a diesel powered mobile vehicle. Engine exhaust is connected directly to the inlet manifold providing primary supply to cool and dilute the toxic exhaust gases.

The Jetflow Airmover will operate effectively at manifold pressures varying from 0.04 bar (0.5 lb/in^2) to 3.9 bar (55 lb/in^2). It is recommended that the maximum manifold pressure does not exceed 4.2 bar (60 lb/in^2). In many applications most efficient operation is attained within the range 0.35 bar (5 lb/in^2) to 1.4 bar (20 lb/in^2).



Coanda's "Lenticular Aerodyne" is a perfect disc, achieving lift by creation of a vacuum around the edge of the wing. Bubble canopy in the center houses the passenger compartment.

Henri Coanda, the inventor of the first jet to fly 46 years ago, has come up with a new concept of propulsion— the "Lenticular Aerodyne".



Lenticular Aerodyne tilts forward for added forward speed.

THE COANDA STORY

By MARTIN CAIDIN

IN THE PENTAGON recently, I was shown design sketches of a disc-shaped aircraft which had high speed, ability to hover, and could rise and descend vertically. This so-called "flying saucer" project has been taken over by the USAF from Avro Aircraft Ltd., of Canada.

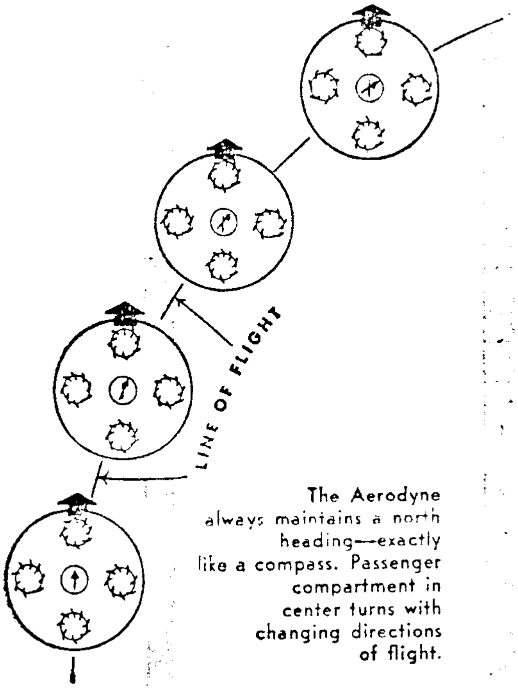
The disc machine achieves its extraordinary performance through a phenomena known as the "Coanda Effect". Engineers have long known, from the work pioneered by the brilliant French Professor, Henri Coanda, that if a jet of gas is discharged adjacent to a surface, the flow tends to follow the wall contour even where the surface curves away from the original jet axis. Jet streams can be induced to turn through angles up to 160 degrees, resulting in a vacuum above a wing shape, with a corresponding high lift from below the wing.

I was told: "If we had listened to Coanda many years ago, and turned our engineers loose to develop his theories, jets would have been with us several decades before now. Coanda actually flew a jet in 1910."

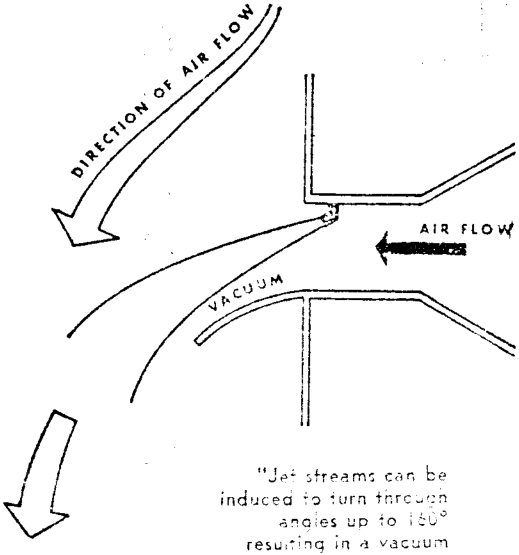
This was news to me. So far as I knew, and according to all the reference volumes, the first·jet flight was not made until August 27th, 1939, when the Heinkel H-178, powered with a Heinkel-Hirth gas turbine unit, reached a speed of 250 mph. But if Coanda actually had flown in 1910—why, then we had had a jet flight *46 years ago!*

The offices of *Air France* in New York arranged a meeting with Professor Coanda, so I could learn at first hand of the unprecedented work this man has accomplished in the field of aeronautical science. Coanda's extraordinary story began, as with so many of the great air pioneers, when he was able to watch the experiments of other men then attempting to develop various models of the new aero machines." In 1904, Henri Coanda witnessed the experiments of the Frenchman, Clement Ader, who seven years before had briefly managed to become airborne in a steam-powered airplane.

After several months of study with his close friend, Ferber, Coanda started to design his own aircraft and, in [illegible] entered the School of Ad-



The Aerodyne always maintains a north heading—exactly like a compass. Passenger compartment in center turns with changing directions of flight.



"Jet streams can be induced to turn through angles up to 160° resulting in a vacuum above a wing shape . . ."

vanced Aeronautic Study. Shortly thereafter, he began his work on the world's first jet aircraft.

From the start, Coanda decided to eliminate the propeller, which he felt had limited potential. His initial calculations indicated to him that he could obtain far greater efficiency by use of a type of aircraft powerplant which he called the "turbo-propulsor."

But let the Professor tell his own story: "From the beginning, I wanted to do away with the limitations of current thinking—that a propeller was indispensable. All airplanes operated on the principle of a kite, and had a propeller either to drag or to push them by motor power through the opposing medium of the atmosphere.

"I believed a more effective motor fence could be utilized by creating a vacuum in front of, or above the aircraft, into which the plane would be drawn. This is a natural phenomenon, demonstrated every time a tornado ravages the countryside.

"I had asked my friend, Clerget, to make me an especially light engine, for which I had given him general specifications. Clerget's inventive genius, in just a few months time, gave me a motor which weighed a mere 2.8 pounds per horsepower, a veritable achievement for those times. It was my intention to use the turbo-propulsor to thrust the air through a circular vent encircling the motor. I intended to inject fuel into the air stream, which would be ignited by the exhaust gases also channeled through the same circular vent. I hoped this would obtain the jet reaction desired, by utilizing the total fuel value of the gas. The exhaust of the ignited gases was directed below and to either side of the fuselage. I armored all sensitive parts with asbestos, and during static testing, failed to note any actual flame at the propulsor exhaust outlet.

"I think a word of explanation is due the special wing design of my jet plane. In order to produce a single piece wing with a plywood-type covering, I lined a large plaster mold with sheets of wood which were covered with glue. To make these sheets of wood take the mold form, I lined them with a layer of celluloid, and then forced hot oil under pressure into the mold. Then I let all the materials cool under constant pressure. But when the time came to remove the wings from the mold after evacuating the oil, I found it impossible to remove the celluloid. It had become so solidly united to the wood that the whole interior of the wing was lined with celluloid.

"The airplane was of revolutionary form. It had a fuselage of slender silhouette, with fuel tanks in the wings. The tail was of delta design, and I even had a retractable landing gear. Most important in the wing design was a slotted flap, which would greatly increase the lift of the airplane. The jet of air was discharged along the upper surface of the flap, and was deflected downward, away from the slot's flow axis. Thus the airflow through the slot stuck to the flap and contributed to the high lift.

"In 1910, the wind tunnel was unknown but I found a way to test the design. I mounted the wings on a railroad locomotive and was able to watch their performance in this fashion. Finally, after all the tests were over and the machine assembled, we prepared for the first experiments. In December, we brought the airplane out of its hangar at Issy-les-Moulineaux and, after a bit of coaxing, started the motor.

"I must admit that I was never a very outstanding pilot. I always seemed unable to shake off a vague apprehension and, that morning, in addition to my usual uneasiness, I was rather excited.

"I climbed into the cockpit, accelerated the motor, and felt the power from the jet thrust straining the plane forward. I gave the signal to remove the wheel blocks, and the plane started moving slowly ahead. I had anticipated that I would not attempt to fly today, but would make only ground tests on the small field at Issy-les-Moulineaux.

"The controls seemed too loose to me, so I injected fuel into the turbine. Too much! In a moment I was surrounded by flames! I had to cut back and reduce my power quickly.

"I worked the throttle and the flames subsided. Only then did I have opportunity to lift my head. I saw that the plane had gained speed, and that the walls of the ancient fortifications bordering the field were lunging toward me. I pulled back on the stick, only much too hard.

"In a moment the plane was airborne, lunging upward at a steep angle. I was flying . . . I felt the plane tipping . . . then slipping down on one wing.

"Instinctively, I cut the gas with my left hand and the jet fuel with my right. The next thing I knew, I found myself thrown free of the plane, which slowly came down, and burst into flames. It was impossible to determine from the wreckage whether the celluloid or the fuel was the cause of the fire.

"But the test was over. I had flown the first jet airplane."

Professor Coanda, despite his bruises, was anxious to begin work at once on a second engine and aircraft. Inexperience with the new propulsor and an unfortunate set of testing circumstances had caused the loss of the initial machine. However, the Professor's funds were exhausted. He had spent more than a million francs of his own money on the project, and could not secure backing. The world had to wait an additional 29 years before another jet airplane took to the air.

During World War I, Professor Coanda applied himself to development of French aviation and allied fields. Many new inventions were fostered, not least of which was the recoil-less cannon designed specifically for Nungesser's fighter.

Coanda, since the end of the First World War, has devoted his attention to perfecting several aeronautical theories which bid fair to revolutionize our concept of the airplane, and its application for personal and commercial use. At this time, in France, he is working on a new personal aircraft design which is a true disc. Imagine—a one to four-place aircraft, of disc configuration, with an empty weight of less than 900 pounds; *a powerplant with not a single moving part;* a cruising speed of 300 mph; and a range, at this speed, of 1,000 miles. Take-off run for Coanda's new aircraft? *None!* It is capable of vertical ascent, descent, and hovering.

If Professor Coanda's estimates are correct—and there is no reason to believe he has erred in his figures—his airplane will be within the price range of the average American automobile, from $2,500 to $4,000. He began research on this newest concept of propulsion, which led to his current design (the correct description is *Lenticular Aerodyne*) in 1930. A student of weather, he saw in the phenomenon of tornados and cyclones a power simile for aircraft.

As Professor Coanda puts it: "Look at a tornado. Its characteristics include a certain kind of wave front, a form of vacuum, with power great enough to ruin entire towns. An understanding of this effect—a vacuum which creates great pressure and lifting power—is what started my research on what has become known as the 'Coanda Effect.'

"Now, most propulsion, such as that effected by propellers, turbojets, and so on, is what I call two-dimensional propulsion. The most important feature of the 'Coanda Effect' is that it is a *three-dimensional* propulsion.

"Take the aircraft wing. In motion the lifting power is derived from vacuum effect on the upper surface, and the greater pressure below. What I wanted to do was to produce the same vacuum, but in circular motion.

"I wanted the vacuum above the surface, and under high pressure, to be derived from a circular motion of gases.

"In 1932, when I was first experimenting with this circular effect, I was producing as high as six tons of lifting power per square yard above a wing. I had created an enormous vacuum above the wing through a propulsion system which, to say the least, was clearly different from anything which had been previously attempted.

"There were a good number of aerodynamicists and engineers who had heard about my experiments, and who wanted to know more. My claims, to say the least, were astonishing and they demanded a detailed paper theory about the entire affair. I had no interest in elaborately-prepared theories; I was more concerned with working results. I had no time to prepare papers for them. So, they couldn't [illegible].

"In 1938 I attended an assembly of the world's leading engineers and aerodynamicists. The people present discussed, among other things, new methods of propulsion, and my friends inquired why I hadn't made a fuss about my ideas. Let me clear up that matter once and for all. Since nobody asked me to address the group, I didn't bother. I had no intention of going to a lot of trouble to shout in anyone's ear.

"I saw many remarkable ideas at this convention. But every single man present was basing his work on the old and proven theories. They were trying to improve the old, instead of striking out in a new direction. They were afraid of being 'different'.

"Now I am a mechanical-minded man, an engineer, and I have been a mechanical engineer from the very start of my career. I am close *to* it, intimate with it, I *understand* mechanical engineering. And that is why I hate it.

"Take our modern aircraft. Things break too easy. There are too many parts, too many things which can go wrong. And when they do, disaster is the result. Just before the last war, I found myself working on a new line of thought. In things mechanical, I reasoned, metal can be brought from its familiar state of a solid to something molten, then to liquid, and finally to a gas.

"Well, why not *start* with gas? Why go through the whole process? Why use the piston engine, which is terribly wasteful with all its heat and friction and thousands of moving parts? Even the turbojet, with its compressors, is too much a deviation from the engine we need. I wanted a machine, a propulsion unit, *where nothing is moving.*

"Naturally, no one agreed with me. This, of course, failed to be of annoyance. I have always been, and will continue to be, an individualist. I don't like to run with the pack, just because everyone has always done so, and because it is the proper thing to do.

"I was well into the new project when —war. With the Germans, well . . . I stopped all my work until 1945."

After the liberation of Paris, Professor Coanda met Colonel Valentine, U. S. Air Attache in the French capitol. He and other officers were sufficiently impressed with Coanda's work to persuade the Professor to leave for the United States, and discuss his work with officials in the Army Air Force. Letters of introduction were thrust in profusion upon Coanda, and he departed for Wright Field, the vital air development center in Ohio.

At Wright Field, the Professor was accorded the welcome of an honored guest. The engineers listened to Coanda, but once he expounded his concepts of wave fronts and aerodynes, he met a sea of familiar faces wreathed in skepticism and disbelief. It was an old story. Coanda was now convinced of the futility of trying to prove that the future in the air lay not in smashing our way through the air ocean by brute strength, but in learning more intimately the nature of movement within the air, and by using to our advantage this knowledge in new designs.

(One example of this "breakaway" from orthodox design is the "area rule" developed by an NACA scientist which produced the so-called "coke bottle" fuselage in the Grumman F11F *Tiger,* the 1,050 mph Chance Vought F8U *Crusader,* and the Convair F-102A. By allowing the air to move along the path of least resistance at the latter half of the fuselage, the F-102A, an airplane which could not attain supersonic speeds and was likely to be abandoned by the Air Force as a failure, suddenly became supersonic even in climbs, *and without an increase in power.)*

Again, Professor Coanda: "It was obvious that I would have to do the job myself. I had a specific project in mind—to show that an airplane, whose propulsion would be derived without a single moving part, would outperform every other aerial machine of comparable size and weight, in the world.

"First, the engine. In my initial experiments I modified the exhaust system of an ordinary piston engine with what I called a 'depressor.' Without touching the engine at all, I increased its power by exactly 53.5 per cent! But the manufacturer rejected the depressor on the ground that it was too complicated, that, if he wanted more power, he would add more cylinders to his present engine. His rejection was not entirely unjustified, for the depressor posed serious casting problems as a production item.

"To come to the present, I am well at work on the new machine, the *lenticular aerodyne.* It is for personal use, designed to hold two people comfortably, or a maximum of four, without baggage. I believe I can produce this new aircraft at the price of the average American motor car.

"The airplane does not, in the sense that we understand it, have any engine at all. The machine functions as a gyroscope. In the air, it maintains a compass' position. That is, it always points to the north.

"Imagine the machine—it is a perfect disc, and in its center is the passenger compartment, with a raised plastic canopy for visibility. Between the edge of the circular wing, and the passenger compartment, there are four large circular openings. Or, if you will, call them depressions in the upper wing surface.

"Each of these openings has along its edge a series of small gaps, through which flame is ejected from a nozzle under high pressure. It is almost like a blowtorch. When the flame moves out between the gap, it is directed away from a funnel intake. This causes a loss of pressure in the funnel area, drawing in a steady flow of air, which is ignited by the combination of air ram pressure and the flame. The flame from the 'blowtorch' we may think of as a continuous sparkplug.

"The burning gases are then blown outward, along the slot. However—instead of letting all the air drawn along with the flame move in the same direction, a baffle on one side of the opening causes the flame to 'bend' in that direction, thus creating a partial vacuum. The greater the force of the flame as it is ejected from the nozzle toward the baffle, the greater the vacuum.

"Thus there is created, along the edges of each of the four circular openings atop the wing disc, a very strong vacuum. Because of the inward movement of air to feed this lowered pressure, the entire circular edge of the disc wing is a vacuum. And, of course, there is corresponding lift from beneath.

"The machine, once it is airborne, never changes its 'compass heading.' It always points to the north, just like a magnetic compass. It is kept in balance by a gyroscope in the center of the lower wing, which is air-driven.

"Turning, or changing direction, is accomplished by changing the pressure (or vacuum) on any one or two sides of the machine. Let's say the pilot wishes to execute a 90-degree turn to the right. By moving the controls, he would decrease power on two of the large circular openings by reducing the power of the flame jets, and the machine would veer off in the desired direction. The wing itself would still point north, but the passenger compartment would revolve to face in the new direction of movement.

"Under maximum-speed operation, the entire angle of attack of the machine would change to tilt forward, just as do the blades of a helicopter. In this fashion, the aerodyne would move forward, backward, to the right or to the left, rise and descend vertically, or hover.

"The safety features inherent in this type of performance are too obvious to elaborate in detail. Every pilot knows the advantage of a machine which he can bring to a halt in midair, and then lower slowly and carefully, if he is faced with adverse weather conditions, or is lost and running low on fuel. And, of course, we have not mentioned the elimination of large airports, or the fact that the machine can truly be landed in anyone's back yard."

Professor Coanda said, in closing. "The United States and France have for many years worked together in aviation," he said, "and it is time we resumed our positions as old friends of the air." End

True, December 1956

The JET-PROPELLED GENIUS and his MIGHTY BLOW

Prof. Henri Coanda, mathematician, engineer and scientist, piloted his own jet plane before you were born and invented the first flying saucer to work. After 50 years of firsts, this remarkable gent is still going strong

By GEORGE SCULLIN

Photographed for TRUE by CURT GUNTHER

PARIS

When Henri Coanda saw his first flying machine in 1908 he was singularly unimpressed. Unlike most of the people in Paris who watched Orville Wright demonstrate the aircraft which was America's newest pride and joy, the 22-year-old Henri could only condemn the Wrights' contraption as a piece of crude machinery that, however new, was unfortunately obsolete.

"It was a kite," he has said many times since, trying to explain that mental flash which makes genius out of mere talent. "It was pulled along by a flailing propeller instead of a length of string, but it was a kite nevertheless. It was all wrong from the start."

So on the spot he devised the jet aircraft. The practical invention of one, though, posed some difficulties, not the least of which was the fact that he knew nothing of flying or flying machines. "Actually, that was a help," he admits now. "I had no fixed ideas or bad habits to overcome."

Such a modest evaluation of his resourcefulness, however, does not quite explain how he invented the first flying saucer, or the first propellerless power boat. Or the scores of other "firsts" he has contributed to our world today. Make no mistake about it—this tall, wise, robust scientist is one of mankind's busiest men and undoubtedly one of the great unsung geniuses of the age.

As for Professor Coanda's jet plane, it took him four years to build. He didn't like the fragility of bamboo and bedsheet wings, so he conceived the idea of wooden wings and then invented the molded plywood from which to build them. When they proved amazingly light and strong, he invented wing tanks to double their usefulness. All told, he produced enough aviation "firsts" to advance aeronautics by several years, and his finished airframe was sleek enough to compare favorably with those of 1956.

The jet engine proved more troublesome. Working on his models was as tricky as de-activating a live bomb, and some of the explosions achieved with appropriate mixtures of gasoline and air would have done credit to dynamite. When at last he had his laboratory models trained to deliver a continuous thrust rearward--instead of forward or through the roof—he still had a fire problem on his hands. Once the fuel had exploded in his single combustion chamber, he could not pass the super-heated gases through the length of a wooden fuselage. He had to release the expanding gases instantly. He did this by providing suitable thrust-pipes around the lower half of his cowling. There remained then only the trick of controlling his flow of power, especially during those delicate moments when his combustion chamber was apt to be pouring out more raw flame than useful thrust. A sensitive throttle control showed great promise of solving this difficulty.

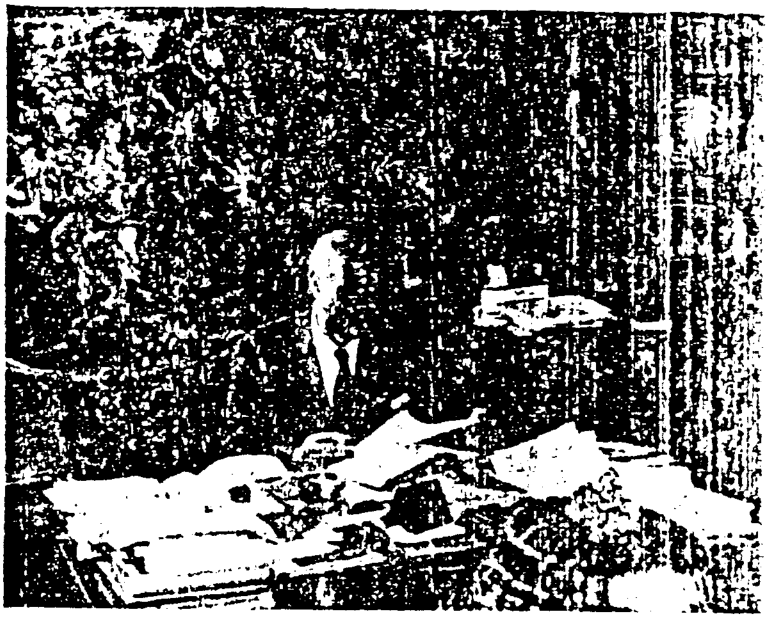
Coanda and his jet aircraft were now ready to learn to fly. Accordingly, he hauled his plane to a small flying field in the suburb of Issy-Les-Moulineaux, just beyond the southwest wall of Paris. In addition to being a small field—airplanes in 1912 could take off and land inside a small ballpark—the Paris wall was still standing as a formal boundary marker of sorts.

As he recounts it, "All I wanted to do was run some ground tests. Check my power for taxiing, check my steering apparatus, check the feel of the thing. As for flying, I did not know how to fly, nor did I know if my plane knew how to fly." Unless overly enthused, his English is always as precise as that. "Besides," he adds, "I was not crazy."

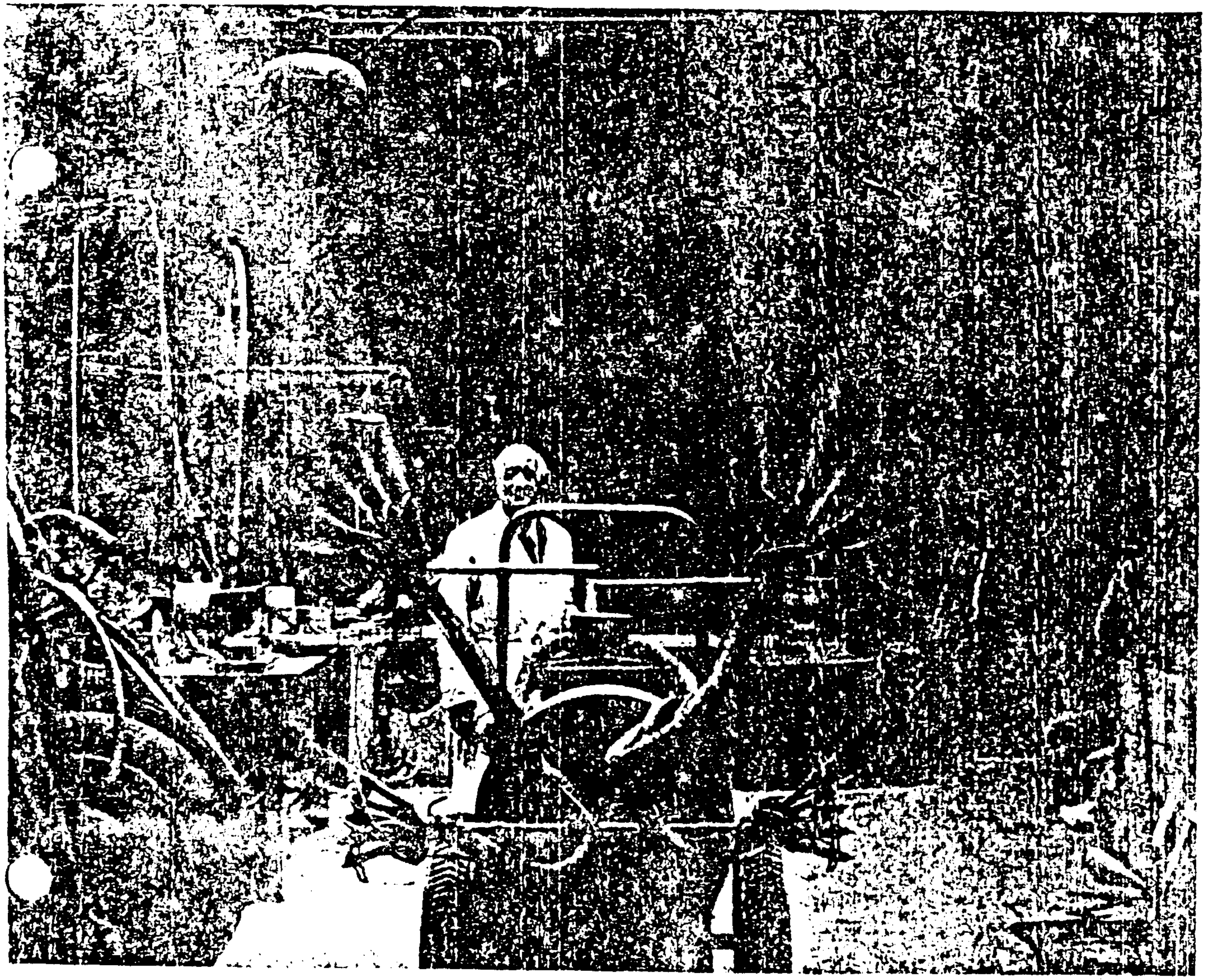
Eye-witnesses of what followed are not in total agreement with that last statement. They saw him climb into the machine at the far end of the field, crouch in the cockpit and apparently adjust the throttle. Then they saw a great sheet of flame and a cloud of smoke. As the smoke cleared they saw the plane flying straight for the Paris wall. They did not regard it the act of a reasonably cautious man.

Prof. Coanda has a more conservative explanation. "I adjusted the throttle, and it marched," he says, the wonder of it all still unlost after 41 years.

It marched indeed. "Apparently I had given it too much fuel," he explains. "When I looked over the side, I saw raw flames shooting out, and that should not be. Not with my wooden wings full of petrol. I ducked back inside to adjust matters. A moment later things felt very different. I looked outside again to find myself many feet in the air. Straight ahead of me was the Paris wall. I did not know what to do,



In his drafty lab (below) or in his elegant home in Paris (above) the genial genius can always be found working.



Coanda's flying saucer research led to this curious by-product—his commercially successful insecticide sprayer.

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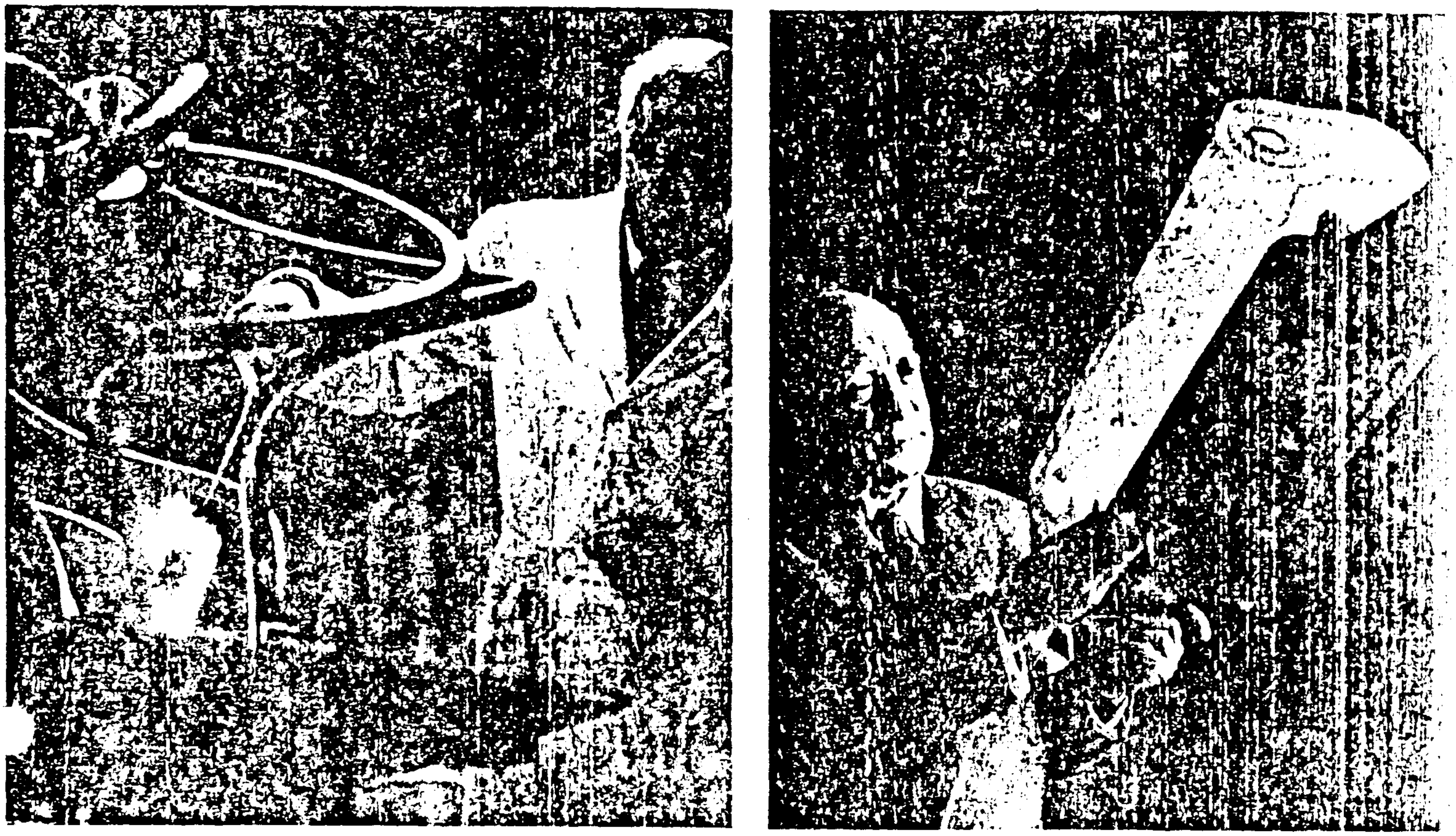
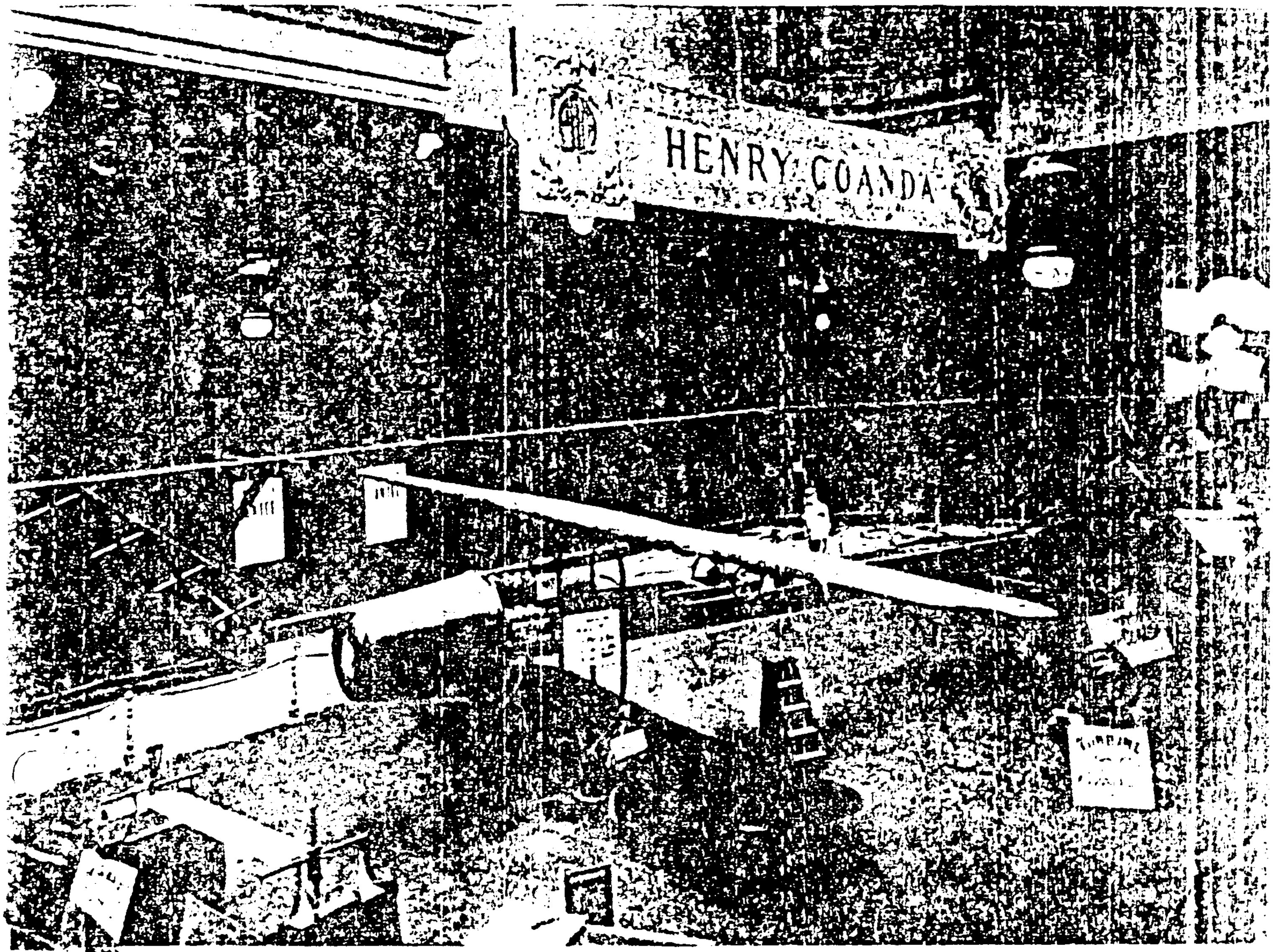


Coanda (left) testing vacuum, and (right) examining model of exhaust gas glow from an internal combustion machine.

The professor experiments (left) with his fuel-oil atomizer. At home (right) he studies a model of his first flying saucer.



Prof. Coanda exhibited this sleek and futuristic jet in Paris in 1909. He cracked it up personally in 1912.

[illegible] fast, and we crashed right at the foot of the wall." [illegible] ection causes him to put his hand to his [illegible] reflectively. "At first I thought I was seriously hurt because my head felt full of a lot of loose things. Then after a bit I felt better because the loose things were not in my head but in my mouth—just a few teeth I had knocked out. I was on my back, and the plane was many feet away, burning up."

Thus ended in 1912 the first jet flight in history, and whether Coanda intended to fly his craft or not, it flew. It was to be 30 years before another jet was to do the same.

The flight, epoch-making though it was, is barely recorded in history. Modest as he is about his triumphs, Coanda thought it best that everyone concerned keep very quiet about the disaster until the situation was remedied by a new, better, and successful jet plane.

There are many versions of why young Coanda did not build a second jet. One story has it that he was broke. Another maintains that with World War I fast approaching, he founded Bristol Aircraft in England and became too busy turning out the famed Bristol fighters for further experiments. Another has it that he suffered a stroke of genius and created the flying saucer, thereby making the jet as obsolete as the Wright brothers' flying machine. And there were other wild stories later—that he had found that plants were thinking creatures possessing hearts; that he was working on a way of irrigating deserts with ocean water; that he was working on a means of destroying the world's harmful insects; or that he was busy in naval architecture with a new driving force called the "Coanda effect," which used neither sails nor propellers.

Simple research discloses that there was no conflict in the stories. They were all true. The U.S. Air Force was investing a few million in a flying saucer based on the Coanda effect. The French government had stopped some Coanda experiments in extracting fresh water from the sea, not because he was not getting a wonderful flow of fresh water, but because he was reaping a rich harvest in a by-product called salt, a government monopoly. Botanists all over the world were using radio-isotopes to study plant life, a process first discovered by Coanda while he was seeking to isolate the heart and brain of a pet flower. The Coanda insecticide sprayer was becoming an international best-seller. In model form the Coanda water jet could be stepped up to boost a model ship right out of the testing tank.

[illegible] matter, or

matters, only added to the confusion. In this age of specialization, how could one man pioneer in so many assorted and highly technical fields? To get the answer, the editors of ARGOSY sent me one cold day last winter to hunt him out in his laboratory in Clichy.

It would be unfair to the tenants of neighboring structures—machinists, garagemen, warehousemen—to say that Professor Coanda's sagging brick laboratory is the worst ruin in the lot, but certainly in terms of modern science, it is as least runner-up to the most poorly housed laboratory in the world.

The only indication that it is a house of science at all is a time-and-smoke-corroded plaque of some dubious metal fastened to an ailing cornerstone. By peering intently at the eroded lettering one can make out: *"Société Anonyme d'Etude Des Brevets et Procédés Coanda. Capital 350,000 frs."*

Inside the sagging door, on which a few flakes of ancient paint can be still detected, the cold and gloom deepen. There is no heating system, and what light there is comes from a few bare bulbs. There is no noticeable improvement in Professor Coanda's large but frigid office.

The first impression is that one has entered a historical room fondly restored by some museum but closed for the winter. The walls, aged a dark brown, are hung with photos, diplomas, scale drawings of forgotten planes and inventions, with here and there a modern photograph looking out of place. In the center of the room is an enormous desk—really three old desks pushed together to hold the mounds of books, pamphlets and papers—and in the midst of the litter is a telephone that looks like one of the original models hand-made by Dr. Alexander Bell. Around the room, in no discernible order, are overflowing file cabinets, old chairs and empty hatracks. No one moves to take off his hat or overcoat.

Then, with a quick step and a warming smile, Professor Coanda enters, and the cold and gloom are forgotten. As for the professor himself—though his passport might claim he will never see 70 again, he looks relatively 20 years younger than he is. Still alert and fit, he has a bouyant air about him that makes the gray in his thinning hair look premature. His conversation and humor are as up-to-date as 1956, or maybe 1957, but with it goes an Old World courtesy one does not associate with modern scientists. He made us feel our request for an interview was a rare and undeserved honor.

"You are not too cold?" he asked solicitously.

It developed that the absence of modern conveniences in the laboratory was the result of a friendly feud between the professor and the city of Paris, owner of the property. The city fathers wanted the site and the adjoining area for some new, income-producing apartment houses. They did not want to throw out the professor and his neighbors—nothing as ungentlemanly as that—but on the other hand they were not going to spend any francs on heat, adequate wiring, or building repairs.

"It is their way of encouraging me to move. They say—to flatter me—I need a laboratory more distinguished."

He led us out on a tour of his current projects, and one glance at the huge, stuffed-to-the-rafters area beyond his office was enough to show why he did not want to move that enormous confusion of interlocking machinery, suspended wind tunnels, swimming pool-sized test tanks, model-building shops, vapor chambers, tool racks of wonderous varieties, testing blocks, and a wide miscellany of objects built for projects long since discarded.

"If we ever have to take this down, it is like that egg in the story. It can never be put together again," the professor said.

In his laboratory Coanda was like a boy with a roomful of toys. Every one of his projects—some going back 50 years or more—held an irresistible fascination for him. Only courtesy prevented his abandoning us in favor of a dust-covered model he suddenly found vastly more intriguing than our company.

"A new idea," he apologized. "I must remember to come back to it." He kept himself well in control until we reached a half-sunken testing tank on the edge of which rested a heavy bronze object shaped like a stunted trumpet.

"A pet," he exclaimed. "You must watch it. It is not new. I have worked on it much, but still I do not know what to call it. I just call it the thing that will make boats run better without a propeller."

At the base or mouthpiece end of the trumpet he pointed out the orifice through which water could enter, and tapped his finger significantly on a wider chamber through which compressed air or live steam could be introduced. Motioning to the bell-shaped end of the device he said, "Watch."

He attached an air hose to the chamber and an eager assistant turned a valve. Though "the thing that will make a boat run better without a propeller" remained motionless, water began to flow into the front orifice, gain strength in the air chamber, and spurt out the bell with amazing vigor. Within a matter of seconds the thrust of the water from the small trumpet was dashing fiercely against the end of the tank. I wished my 25-hp outboard motor could do as well.

"Enough," said the professor.

Either through mistake or uncontrolled zeal, the assistant turned the valve wide open. The tank half emptied itself over the wall. Water flowed on our shoes. The professor smiled. "It washes floors, too."

On the way back to his office he stopped at a bright red commercial version of his insecticide sprayer. "There is no poison in it," he observed. "Only water."

Once more his all-too-willing assistant started a motor. Nothing seemed to come out of the sprayer but a rush of air. Then, slowly, a fog bank began to appear at the far end of the laboratory. It swelled and rolled over us with the suddenness of a fog taking San Francisco. "Enough," said Coanda. The motor cut out, but the fog remained dense and impenetrable.

"It's the Coanda effect," explained the professor. "It shatters the water. Atomizes it, as you say. It will hang around for many minutes."

We groped our way out. Inside the professor's office the door had to be closed—forcibly, it seemed—against the pressure of the fog that sought to follow us.

Unlike most geniuses, who have a marked affinity for being born in poverty, Coanda demonstrated a prenatal astuteness in being born to wealthy, indulgent parents in Bucharest. In return for his good fortune, he became first an infant prodigy and then a child prodigy in higher mathematics. By the time he was 16 he had outgrown his local professors, and his admiring parents sent him to consort with such great engineers as Dr. Alexander Eiffel, of Eiffel Tower fame, and such great mathematicians as Dr. Paul Painleve, so admired that three times he became Premier of France by what amounted to popular acclaim. Under their sponsorship Coanda's scientific stature grew so fast that by the time he was 20 he was in a fair way of following Einstein and a few other physicists into the abstract realm of pure mathematics.

An intensely serious scholar, packed with energy, he plowed into time-hallowed scientific theories like a pin-shattering bowling ball. He agreed with nothing. The more sacrosanct the theory, the less he agreed with it. Suddenly he found himself in a violent feud with the internal combustion engine.

To Eiffel, Painleve and hundreds of great engineers, the internal-combustion engine was the great god of the twentieth century. It was the machine that would revolutionize industry, save the working man, and as an added dividend put a gasoline buggy, naphtha boat, and flying machine in every home. Coanda thought it stank. He still does.

"How," he asks reasonably enough, "can you perfect a machine that commits suicide with its own friction? It is full of moving parts, and as an engineer, I resent moving parts. You will see that in fifty years the internal-combustion engine will be recognized as a freak toy, a wasteful, inefficient piece of folly. You will pardon me if I speak heatedly, but the world would be much farther ahead if engineers had seen that fifty years ago." Modestly he fails to add, "As I did."

Nevertheless, Coanda remained discreetly silent for some years, partly because he was too courteous to throw rocks at the idol of his personal friends, but mainly because he detected a certain weakness in his logic. According to the engineering logic of the day, if an automobile, ship, airplane, or even a stationary engine moved or lifted parts from one place to another, then certainly it was inevitable that parts had to move. Until 1908, Coanda found no way of getting around that.

That was the year Dr. Painleve, serving one of his periodic terms as premier, combined his admiration of internal combustion engines and flying machines by inviting the Wright brothers to demonstrate their airplane in France.

"And pouf! there it was," recalls Coanda. "A beautiful airplane anchored to s internal-combustion engine with a at big whirling club flailing the air into submission." He shudders. "Air doesn't need all that abuse. It is made of gases, and needs no more than gases to overpower it. Look, if you blow up a balloon and release it, it jets off very fast. That is what I thought then. 'All you need,' I told myself, 'is a constant flow of gas from the aircraft like that escaping from a balloon. Surely that cannot be too hard to arrange.' "

It was arranged, with his jet engine containing no moving parts. "What happened after the crash?" we asked. "Why didn't you build another jet?"

"It is a sad story. Very sad. Somebody talked."

The talk that did the damage reached no war departments, no aircraft manufacturers, no kings or potentates. It reached only his father, which was enough. Up to that moment his parents had been footing the bill for his "scientific experiments," still fondly hoping that their son would find himself and become a professor, a position regarded in Europe as eminently respectable. When they found young Henri was throwing away his patrimony on what anyone could see was an expensive form of suicide, they summoned him.

"From now on," said his father, "you are on your own."

For the first time in his life Coanda was forced to think about money and where it came from. He found it dull and unrewarding work. To this day money is something he ignores as much as possible.

His first commercial venture was to try to sell France on his jet aircraft. When the authorities discovered his plane consisted of a small mound of ashes just outside the city wall, he was made aware of the futility of pressing the sale. He moved on to England. To his complete amazement the English bought not his jet engine but his airframe. So many investors were so impressed with his molded plywood wings, his wing tanks, flexible landing gear, simplified controls and a multitude of minor improvements that the next thing he knew he was founding Bristol Aircraft, soon to become one of the leading aircraft manufacturers of the world.

By 1925 Coanda, the scientist, was bravely facing the grave dangers of riches and success. He was saved only because his feud with the internal-combustion engine was blazing as strongly as ever. He had been too successful in adopting his airframes to conventional motors, and with the whole world sold on piston-driven propellers, his board of directors saw no future in wasting money on Coanda's pet theories.

His intentness drove him to suicidal lengths. To carry on his research in gas combustion, airflow, lift and a mysterious thing called vacuum, he taught himself how to fly. Now it is one thing to possess the creative genius that can design a complex and advanced aircraft, and quite another to possess the physical coordination and daring to fly it, a fact that has kept many designers living a lot longer than many test pilots. Coanda brashly decided they were one and the same thing.

"I was a terrible pilot," he admits now. "A most terrible pilot. Formidable."

Though Coanda has always been given to understatement, those who have endured the horror of watching him fly consider Coanda's description of his flying ability to be gross understatement.

successful landing, it was a case of plane meeting runway through sheer coincidence."

A typical flight, for instance, would find him cruising at dangerously slow speeds between ripe grain fields and dark forests, checking not only the thermal differences but the differences in the ionization of the air over light surfaces and dark. Then he would drift over to loiter around a growing cumulus cloud, an aerial Peeping Tom trying to discover the moment of conception that created a droplet of pure water out of invisible gas. In the meantime he would be making copious notes on his observations, jotting down instrument readings, and occasionally remembering an old flying rule that if he was hanging by his seat belt, he was probably upside down.

Theoretically his copious notes were destined to augment his laboratory research on jet aircraft, but the conclusions he reached had an annoying habit of dead-ending in the midst of nowhere. Nevertheless he stored them all in the back of his mind. They made quite a clutter, seemingly unrelated and harmless.

Then one day he was flying along when he was struck by a second spark of genius, and a whole clutter of ideas erupted. In that flash his own aircraft became so obsolete that he was surprised to find it still able to fly. Gingerly he returned to the field, hoping the old laws of aerodynamics would last long enough to get him safely down.

"You feel very good when something like that happens," explains Professor Coanda. "You don't know what you have got, but you feel good anyway. Light-headed, and you say to yourself, 'Tomorrow the details will all come into place, and you will know what you have, and it will be wonderful.' "

If he didn't know what he had, he knew what he was going to get. As on the day he had condemned the propeller-driven aircraft of the Wrights as obsolete, and invented the jet, he now condemned his pet jet project as obsolete, and invented the flying saucer. Small wonder geniuses are apt to be called crazy. With the jet age still a quarter of a century in the future, here was a brash young man not only declaring the jet obsolete, but proposing to prove it.

And therein lies the difference between Prof. Coanda and his academic colleagues. Coanda has always shunned the ivory refuge and personally taken on the jeering, practical world. With his flying saucer, he gave it plenty to jeer about.

The Coanda effect did not arrive full grown in Coanda's mind that day. He was merely watching airflow ribbons prove what the birds, the bees, and the plane-designers had long since discovered: that when a curved wing or surface is passed swiftly through the air, it is lifted not so much by air pressure beneath the wing as by a partial vacuum created above it. Coanda now found himself regarding this time-honored process as a most awkward procedure. Why pass a wing through the air? Why not just pass the air over the wing? In either case,

The first laughter came from aircraft designers and manufacturers. They pointed out there were only two ways of passing air over a motionless wing. You could place a captive plane model in the air stream of an electric fan, or you could cage it in a wind tunnel. In the first instance, you would have difficulty in keeping an electric fan suspended some distance ahead of a full scale plane; and in the second instance, free flight would be even more difficult in a wind tunnel.

Coanda, however, was not thinking in terms of fans, within wind tunnels or out. He was still thinking of his jet engine, but in a vastly different way. He had thought of his original jet in terms of thrust. Now he thought of it in terms of lift. The first had forced the wing through the air to get lift; the second would force the air over the wing, and the lift could be vertical, horizontal, or in whatever direction desired.

"All the while I was thinking this, I was going backwards," he says cheerfully. Plainly he recalls those frantic, harried years with pleasure. "When I built my jet airplane, I knew nothing about aircraft, and had to learn from the bottom. Now I was a good aircraft designer, quite good, and I had to unlearn from the top. It was vexing."

In going backward he shed first his wings, or at least the conventional theories on wings. Then he shed the tail, and finally he eliminated the fuselage. Along about 1980 his design was down to an irreducible minimum. A disk as shallow and as empty as a clay pigeon.

Coanda wanted his aircraft, which he had now dubbed the renticular aerodyne to rise vertically. A conventional aircraft could only rise vertically if its airspeed was slightly in excess of its headwind. An airplane flying 110 mph, for instance, could head into a 100-mph wind, and convert its excess speed into vertical climb. That took a lot of power on the part of the plane, and there was the added difficulty that 100-mph winds were not always conveniently available at all airports. Elsewhere the concept of vertical take-off was getting some small support from helicopter enthusiasts but they, too, were experiencing difficulties. In business could prove to each other that the helicopter could fly, but they couldn't prove it to the helicopter, which remained obdurately on the ground for another nine years. Besides, Coanda looked upon the helicopter with scorn. It was full of moving parts and flailing rotors.

His own problem was the simple kind that drive men mad. On paper he could prove that his renticular aerodyne, when activated by a self-generated flow of air over its surfaces, should rise like a sky rocket. But in actual tests with models he could produce only a whistling noise with his airflow. Nothing else happened.

A pipeful of smoke offered the first clue to what was wrong. One day he was watching a stream of air flow over a platter-sized disk, producing a lift that could theoretically tear the wings off a conventional plane, when some tobacco smoke was drawn into the current. For a fleeting second he saw in the swirling smoke a strange pattern. At once he was reminded of the ribbons he had used in studying the airflow over the wings of his old Bristol. Ribbons were too coarse to trace the minute disturbances of air flowing over a model, but smoke. . .

In the next few days Coanda ate more smoke than a firefighter does in a lifetime, but he had the answer to his problem. According to his theory, the Coanda effect was produced when air created a vacuum in passing over a curved surface. If he used a smoke bomb to charge his air chamber, then where there was smoke there had to be atmosphere. Where there was no smoke, there was vacuum.

What had thrown off his mathematical calculations was a little thing that only in the last few years has assumed importance in aerodynamics. A thing called the "boundary layer"—that thin skin of air which clings almost magnetically to a surface even at super-sonic speeds. That added molecular layer of air bubbling on the surface of a model had been enough to distort the theoretically smooth flow of air. As a result, the vacuum Coanda had calculated was producing a lifting effect above his aerodyne could now be seen (thanks to the smoke) trailing ineffectually at some distance to the rear. A slight alteration of the direction of airflow brought the vacuum back where it could do some good.

But Coanda still had another enormous hurdle to jump. No power. "You will recall in 1934 the airplane engine was becoming very good" says Professor Coanda, but he cannot resist adding, "for powering kites. My poor old jet engine of 1912, it would no longer do. Certainly it would not do for the model I intended to build. For a large aerodyne, maybe yes, but I was mindful that my record as a pilot was not of the best, so I was not tempted to build a large one to fly myself. I was alive after much, and was quite happy to stay so."

The power source Professor Coanda finally devised was clumsy but effective. Instead of using compressed air in small cylinders, he now installed a large air compressor. Instead of building one aerodyne model, he built two, and in them installed the unique Coanda baffles that downward over their top surfaces. ("Backward" is a direction applied only to the models. In his present experiments, Coanda is working on controls that will apply his effect in any direction and at any speed.) To the bottom of each aerodyne he attached bullet-shaped reservoirs for his compressed air, the reservoirs being regulated to keep the flow of air constant regardless of the pulsations of the air compressor.

All that remained to do was work out a suitable connection between the air compressor and the air reservoirs that would permit the aerodynes enough freedom to move. The device he hit upon is strikingly similar to that used today by jet-powered helicopters. Just as the jet pods are mounted at the ends of the rotor blades, his two aerodynes were mounted at the end of a hollow tube; where the pods are supplied with fuel through a connection at the axle, his aerodynes were supplied with compressed air. In fact, so identical are the two arrangements that if one wants to be old-fashioned enough to think in terms of helicopters, Professor Coanda is convinced the use of miniature aerodynes at the tips of rotor blades would be far smoother and more efficient than the use of jet pods.

Once his aerodynes were properly aligned on the hollow tube, and the tube mounted to revolve freely on its heavy stand, all he had to do was turn on the compressor. Which is like saying that once the mountain is there, all that remains is the climbing.

Coanda turned the valve.

At first there was only the shrill whistling of escaping air. As far as the Coanda effect was concerned, the air might just as well be blowing out of opposite ends of the hollow tube. Then the whistle turned into a snarling whine. From each aerodyne the ejected air began ripping holes in the atmosphere, leaving gaping voids of vacuum. Drawn into these voids were the aerodynes, faster, and yet faster. Spining around their axle they became a screaming blur.

It was something to see. Two aerodynes spinning on an axle as though they would fly off into space, heavy stand and all, and impelled in their furious orbit by a power never used by man before. A mighty power made up of nothing at all—a vacuum. The air compressor pumped away, and the compressed air screamed past the baffles in a din that was strangely comforting. At least one could understand the familiar power of man-made compressed air, and even understand its use in jet propulsion. But what staggered the imagination here was that the aerodynes were approaching the speed of sound not through jet propulsion but in defiance of it. The air was jetting off into space at right angles to the line of flight, leaving behind a void. That was the awesome thing—the disks were being drawn into a void. Where might they go in free flight?

At the end of the first successful demonstration, Coanda and his assistants were stunned, first by the silence that followed the last sigh of air, and then by the significance of what they had. Nearly

tween the first conception of the Coanda effect and the laboratory proof that it would work. What lay ahead was an aircraft so revolutionary in concept that it would make obsolete every airplane then flying in the world.

"We had the figures to prove it," Professor Coanda says. "On paper we had created a passenger-carrying renticular aerodyne that could rise vertically, hover, fly faster than the speed of sound, and land on a surface no larger than itself. We used those same figures to build our laboratory models. When our laboratory tests proved successful, we knew there was no scientific reason why a large, passenger-carrying model could not work. True, we still had to invent a new source of power to provide a high-speed flow of gases, and we had to invent steering controls that would provide complete stability from zero to several hundred miles per hour, but those were mere details. The main thing was that the renticular aerodyne worked!"

Aware what goes on in the cold, realistic world of commercial aviation, Coanda hastened to protect himself from the hordes of aircraft manufacturers who soon would be storming his doors. He took out detailed patents on the aerodyne and the Coanda effect, not only in France but in England, the United States, Germany, and other technically-minded countries. In the light of subsequent flying saucer stories, this action becomes important. In the mid-thirties, before a security lid was clapped on scientific advances in strategic fields, all patents pertaining to aviation were carefully scanned by all aeronautical engineers worthy of their hire, and Coanda's applications came complete with diagrams and sketches, with nothing concealed. He sat back to await the rush.

He didn't even get a postcard.

Once more, as in the case of his jet airplane, he was too far ahead of his time. As far as the commercial world was concerned, he might just as well have proposed another wild scheme to sell perpetual motion. In the academic world, however, his stature was such that he received slightly more flattering attention, notably in France and Germany. In response to his learned and detailed scientific paper on the Coanda effect, German physicists did him the courtesy of building scale models of his renticular aerodyne much like those he had used in his own early tests. They reported back that they worked, and made amazingly successful classroom toys, their shrill noises keeping even the most dormant students awake.

Here again, in the light of flying saucer developments, an innocent action becomes important. In the course of World War II the Germans revived Coanda's theories on jet propulsion successfully enough to throw quite a scare into Allied pilots, but they unleashed no flying saucers. That does not mean, however, that the saucers were lost or forgotten. They may have been well along in development when the Russians swooped in to clean out the research centers, scientists and all. That part is speculation, but Coanda is convinced that among the scientists who became hurried guests of Moscow were several familiar with his effect. And he is equally convinced that the disks recently seen hurtling through the skies of northern Europe are the products of their handiwork. Certainly the Unidentified Flying Objects program of the U.S. Air Force does not discount that possibility.

Mystery surrounds German developments of the aerodyne, if any, but the same cannot be said of what the French physicists learned prior to World War II. In 1939 Prof. Albert R. Metral, chief of the Technical Cabinet of the French Air Ministry, completed a series of exhaustive tests on the lenticular aerodyne, and published his findings in a scientific paper that received world-wide circulation. Perhaps the most astounding item in the lengthy report is the revelation that when the Coanda effect takes place over a lenticular aerodyne, the resulting lift is equal to a staggering two-and-a-half tons per square yard of surface. In short, enough lift to permit a vertical take-off in any lenticular aerodyne not made out of lead.

The U.S. Air Force has now revealed that it has a few million dollars invested in a saucer-shaped aircraft based on the Coanda effect, being built by AVRO Ltd., of Toronto, Canada. And now can be revealed one of the best-kept secrets of aviation history—a matter of public knowledge for more than 20 years. Like Edgar Allan Poe's story of the purloined letters, best concealed when it was most conspicuous, Coanda's flying saucer completely vanished in patent office publications, laboratory experiments, and scientific papers published by the French Air Ministry.

More amazing than the case of the purloined saucer is the next step in the development of the Coanda effect. Unable to swing the full-scale development of the aerodyne with his own dwindling supply of francs, Coanda was forced to suspend research in that direction and hunt around for some other and more remunerative aspect of his effect. He found it, ironically enough, in the internal-combustion engine, that hated contraption filled with moving parts that had driven him to invent the jet engine in the first place.

Second only to his hatred of moving parts in an internal-combustion engine was Coanda's loathing of its wasteful inefficiency. It struck him as ridiculous, for instance, that exploding gases in a cylinder were at peak power when the piston was at the bottom of its stroke, and that it was against this peak of power that the piston had to return. The exploding gases were not drawn out of the reluctantly opening exhaust valve; they were forced out, fighting every inch of the way, by the scouring stroke of the piston. The loss of power was enormous. "So I say to myself that the Coanda effect will take place wherever gases flow under high pressure. Next I ask of myself, 'If I apply the Coanda effect to the exhaust gases of an internal combustion engine, what good will it do?'" He shakes his head with a wry grin. "So I thrust upon a defenseless world another internal combustion engine."

The Coanda engine, with an efficiency rating 40 percent higher than that of conventional two and four cycle engines, has yet to set the commercial world afire. But it did serve to keep his laboratory out of bankruptcy, and this in itself may still prove of more value to the world than a few million francs in cash. What the unique engine does is replace the exhaust valve with a Coanda effect baffle, and substitute for the labored scouring stroke of the returning piston, a vacuum to draw off the exploded gases. The technical aspects can be explained only in several pages of complicated mathematical formulae, but the net result is a constant flow of power-producing explosions freed of all back-pressure from the exhaust. Coanda does not blame any conspiracy of manufacturers for the lethargic reception accorded his engine. He is delighted that anything as obsolete as an internal combustion engine, even his, could do as well.

His next venture was a surprisingly logical one. If the Coanda effect could eliminate the propeller of an airplane, why could it not do the same for a boat? That there were some basic differences between boats and aircraft and between water and air, Coanda happily chose to ignore.

Water, he has found, is the most accommodating of all gas combinations, and the most rewarding to those who care to pry into its secrets. Its two parts of hydrogen to one part of oxygen perform readily to changes of pressure and temperature, 180 degrees being enough to convert solid ice into liquid and then live steam. It is the major part of all living organisms, and hence the main support of life itself. It comes in ocean lots or invisible droplets without loss of integrity, and will obligingly run up hill or down. It is a good mixer, behaves logically most of the time, and yet cloaks itself with a mystery that keeps it ever fascinating. Most mysterious of all are its sources of tremendous power, few of which have been tapped.

It was with the conviction that the Coanda effect could tap one of these mysterious sources of power that Coanda began his research. His experiments led him far afield—he became an authority on the sex life of aphids, organic fertilizers, radioisotopes, and the mental powers of vegetables, among others—but his straying never took him far from the Coanda effect.

Obviously, if he was to study the movement of water, he first would have to learn why water moves. Gravity would explain why it moves down hill and differences of water temperature would explain some of the ocean currents; capillary attraction would, or should, explain why water moves up hill. The trouble with capillary attraction was that it offered only a weak and partial explanation. In laboratory experiments with fine tubes he could use capillary attraction to lift water several inches. But if botanists were correct in assuming that plants owed their healthy circulation to the same principle, then the 60-foot trees in the backyard were doing a much better job. How come? What did the trees have that he couldn't make better?

Coanda went into the plant-growing business to find out. He watered his protegés, and discovered that water did indeed flow to the topmost leaves, not a particularly new discovery. To advance fundamental research, he had to discover just how, why and when a specific drop reached its ultimate destination. That was a particularly difficult assignment since one had yet discovered how to tell the difference between one droplet and another in a pitcher of water. Coanda was nonplussed only as long as it took him to ransack the attic of his well-stocked mind. There he found a stray bit of information on isotopes and at once conceived the idea of tagging his droplets with radio call letters.

During the following weeks he traced his isotopes as they were absorbed by the roots of a plant and journeyed upward through stalk, stems and leaves. To his astonishment, they did not move steadily as in his laboratory tubes. They moved according to a definite rhythm. They moved, paused, and moved on again, almost as does a blood cell responding to a heartbeat. Intrigued by the curious pulsations, he hunted for their source, and while admitting that he has yet to isolate any vegetable organ that functions exclusively as a heart, he does claim that the pumping action originates in the complicated area in which stalk or trunk merges with the roots.

"The brain is there, too," he asserts blandly, his smile denying any touch of the crackpot in his unique ideas on vegetable intelligence. "Plants do have a brain, you know. How else can you explain why some plants will turn with the sun, or store water against droughts, or climb walls, or attract insects for sexual fulfilment, or even trap insects for food? Normal physical laws cannot cover all of the hundreds of activities plants enjoy."

An assistant points out an interesting sidelight on Coanda's discovery of plant intelligence. "As soon as he began to suspect plants of being thinking creatures, dim-witted though they might be, he began to worry about their welfare," says the assistant. "He puttered around their pots, feuding with all the insects that threatened them and getting indignant over fungi or blight. Then he started wondering about their diets. We were feeding them the best of fertilizers, but that was not good enough for the maestro. He had to know what each preferred personally, and soon he ordered that we no longer use inorganic fertilizer. He claimed the plants had no taste for it, and ate it only when there was nothing else around."

We brought the subject up during an interview and found him as adamant today as ever. "Out of life comes life," he asserted. "Inorganic fertilizer, having no life to begin with, has nothing to add to life. Only organic fertilizer can pass on the life force, as I am sure any plant would tell you if it could only speak."

More was to come of this odd sideline, but in the meantime Coanda had picked up a major clue on how to apply the Coanda effect to boat propulsion. His studies had taught him a lot on water as mass, on water as a moving liquid, and water as an individual droplet. Now these assorted studies and scores of other hitherto unrelated facts began to fuse in one of those bits of intellectual fission that is called a stroke of genius. When the sparks cleared, what he saw first was a ship with a large orifice below the waterline of its bow into which the water flowed. The ocean represented water in the mass, or resistance, and the water entering the orifice represented a liquid in motion. Any application of power that would speed up the flow of water through the orifice, and thus naturally speed up its ejection from the stern, would serve to overcome the resistance of the static water mass ahead. Even a propeller powered by moving parts could do that.

But compared to air, water offered such enormous resistance that the production of an underwater vacuum seemed virtually impossible. Then Coanda, who knew next to nothing about ships to begin with, was informed of a curious behavior pattern common to all ships at sea. They pitched.

Immediately his imagination projected a horrifying picture of a Coanda ship lifting its bow over a wave. Into the orifice would pour, not solid water but thin air. The least that could occur then would be a burp in the workings that might well pop some rivets in the hull. The worst would be a side-splitting belch that would sink the ship. At once Coanda abandoned the orifice-nosed ship, "fastening my life jacket as I went."

He was thinking fast even as he abandoned his projected ship. Why one orifice in one bow? Why not many little ones along the keel, or, in small boats, a couple under the stern?

In its present stage, the result is the "thing that makes a boat run better without a propeller," already described. It uses the Coanda effect, but the marine version otherwise bears no resemblance to its aerial contemporary, nor does it operate in the same way. The vacuum that produces *lift* over the lenticular aerodyne, for instance, becomes *thrust* in the trumpet-shaped "thing." And where the gases rip over the *outside* of a saucer, they swirl through the *inside* of the thing, the Coanda effect taking place over all 360 degrees of the curiously-shaped air chamber. The vacuum created in the chamber thus exerts a force that is even more concentrated than that created in the free air above an aerodyne, and its effect upon water is shattering. In the model powered by compressed air that we watched, the first thing we noticed was the way the water was drawn through the front orifice as the Coanda effect began to form in the air chamber. It looked as if the vacuum was drawing it into the chamber all right, but from that point on it looked as if the compressed air was blowing it out through the trumpet. The thrust appeared to be no greater than one could get by submerging the compressed air hose alone.

Then the pressure was stepped up. Water was drawn into the orifice in a mighty, continuous inhalation. Vacuum seized it with explosive force. Shattered to molecular size, its volume enormously enlarged, it was seized by the compressed air and ejected through the trumpet. Swollen to even greater size by the air mixture, it blasted into the tank, a midget of water grown into a rampaging giant of thrust.

"That is with the small air compressor, of course," explained the professor. "When we use a large one, something else happens."

What happens makes still another story of the amazing Coanda effect. One day, wishing to see what the thing would do under really high pressure, Coanda hooked it up to a giant compressor and gave it the works. It bored a post-sized hole through the water the full length of the tank, and then sent a tidal wave over the walls. Curious about the hole, and wanting to examine it in more detail, he up-ended the thing at an angle, with the bell of the trumpet sticking slightly above the surface of the water. Once more he turned it on.

He got a nice fog-horn bellow out of the trumpet, but of the hole there was no sign, not even of its emptiness. Then, strangely enough, the hole slowly began to appear, looking, however, more like the ghost of a hole than the hole itself. So astounded was Coanda by this psychic phenomenon that some moments passed before he was aware that the whole laboratory was densely packed with fog, and the thing was merely blowing a hole through it.

The performance might have been dismissed as just another trick in the repertoire of the thing had not Coanda been in the midst of his feud with the insects attacking his pet plants. "When I saw where the fog was coming from, at once I say to myself, 'This thing will spray a lot of potatoes.'"

There may be some irony in the fact that the first flying saucer to achieve a gratifying commercial success is to be found in the heart of a potato bug sprayer, but that is where it is. Coanda was thinking more of plants than he was of francs when he developed his first Stromboli sprayer, which can send a fog of insecticide over a wide area or into the topmost branches of tall trees. The droplets, being so fine as to be almost invisible, respond to ionization, and are drawn almost magnetically to all leaves, coating them top *and* bottom. As an added feature, the microscopic droplets settle themselves snugly into the minutest cracks and crevices of the leaves where they cling tenaciously even through a heavy rain.

As soon as word of what he had reached the farmers and fruit growers, a spontaneous rush began. Today orders flood in from all over the world, and with insect-plagued, impatient customers demanding immediate delivery from a production line already months behind, Coanda nearly foundered in success. Fortunately his assistants were able to rescue him with a bit of strategy. Reminding him of the good old days when he had nothing to do but research, and no money to do it with, they pointed out a couple of projects that had run aground on the shoals of insolvency. Now enough francs were flooding in to set them afloat once more.

With a sigh of relief, Coanda dropped his unfamiliar role of a business tycoon with money and worries, and began to play with mirrors. Once more he was far afield without actually leaving home. The mirrors had no part of the Coanda effect, yet it was his study of the Coanda effect that led him to the mirrors. To be specific and logical about it in a Coanda sort of way, it was the study of the Coanda effect under water that led to his mirrors in the sun. That and his curiosity about how clouds were born.

Once he had been concerned with the willingness of hydrogen and oxygen to give up their free state as gases and wed themselves polygamously, two hydrogens to an oxygen, as water. Given a few changes of temperature, and they would separate just as willingly. Other chemical compounds, turned into gases by high temperature, always seemed averse to reunion when the temperature was reduced, but hydrogen and oxygen could usually be counted upon to reappear as water. What was this strange affinity?

Now he thought he knew, but it didn't make sense. He began with the premise that warm air can hold more moisture than cold air, though in either case the moisture is not there at all, having turned into more air or less air. Refusing to be confused by airy or airier air, he established as his next premise that warm air seeks moisture, and the warmer it gets, the more desperately it seeks it.

Testing out this premise, he found that if one just took a few cubic feet of ordinary breathing-type air, superheated it, and then thrust a wet blotter into its midst, the moisture vanished as though inhaled by a vacuum cleaner. The blotter could then be withdrawn without being warmed up enough to raise a sweat. He had now established the long-established fact that evaporation produces a chilling effect, all of which is why humans have sweated on hot days for many thousand years. But now Coanda went a daring step farther. He asked, why?

Once more we will omit scores of pages of complex mathematical formulae, and come up with an enormous, silo-shaped affair, the walls of which are made up of mirrors to reflect the sun's rays. When the silo is centered just right on its axis, the sun is bounced from mirror to mirror so often that it becomes hundreds of suns, and what one hot sun can evaporate from a given body of water, the silo can do a thousand times over. When appropriately mounted over a tidal pool, the bottom clearing the water by several inches, the silo heats the air within itself to blast furnace intensity. The hot air, behaving normally, rises, and because it is superheated hot air, it rises with a volcanic rush. The rest of the performance does not behave in so normal a fashion. To replace the hot air rushing out of the top, cold air is drawn in at the bottom. That is the point to remember. The air that is drawn over the surface of the water and up into the silo is plain, ordinary cold air, just like all the rest of the air outside the silo. Not until it gets well along on its upward journey does it become heated enough to support abnormal amounts of moisture. Yet by this time it is loaded, and becomes more so the higher and hotter its gets. It is super-saturated by the time it reaches the top where it is drawn off by fans to the condenser. The amount of water that then flows from the condenser is a source of astonishment to all who have seen it, including Coanda. The tidal water, in the meantime, does not even change temperature, and were it not for the accumulation of salt beneath the silo, one could not believe a concentrated evaporation was taking place at all.

The salt deposits have put the solar water pump on the shelf temporarily. Coanda thought he was perfecting a way of drawing fresh water from the ocean and leaving the salt behind; the French government thought he was perfecting a way of drawing out the salt and throwing away the water. This looked like a serious threat to the government salt monopoly, so until some legal decision is handed down on just which end of the pump is in operation, Coanda has turned to less harassing but more scientific problems.

As of the moment he is working on some 40 or 50 projects, some of which are bizarre enough to show promise, but in the main he is occupied in developing a system of controls for his lenticular aerodyne. He wants to do away with the use of conventional steering vanes which might yield under the pressure of high speed maneuvers, and use the Coanda effect itself in place of ailerons and tail surfaces. Through a series of ports around the flying saucer, he hopes to give the pilot perfect control of his aircraft in all directions, not only the conventional up-and-down and left-and-right, but for braking and reverse as well. With no surfaces exposed outside the aircraft, the only moving parts needed would be those required to open or close the desired ports, and as he sees things now, these could operate from a single steering column no more complicated than an airplane stick.

Oddly enough, in contrast to the public conception of the flying saucer as [illegible] for the moon traveler, Professor Coanda does not see his aerodyne as [illegible]. "That is nonsense," he says without the usual twinkle in his eye. "Up in the stratosphere you need all sorts of uncomfortable pressure and oxygen things to protect your passengers. I say the lower altitudes where your passengers are comfortable, where there is plenty of oxygen for your combustion chamber without all sorts of whirling superchargers, and where the atmosphere is thick enough to give the Coanda effect a firm hold."

But what happens when you fly at high speeds through rough weather? Won't it shake you apart?

The twinkle comes back. "You are thinking of kites. Conventional aircraft are all subject to turbulence. But you must remember that in the lenticular aerodyne we create our own vacuum, and in a vacuum there can be no weather. Just smooth sailing all the way."

One last question. When will the flying saucer age arrive?

Professor Coanda looks astonished. "Arrive? Why it is here now."

We left with the deep conviction that if Professor Coanda says it is here, it's here. He's the man who made it.

—George Scullin

Back in the dim days when
airplanes were crude aeronautical
experiments, he invented
a jet plane, then
a flying saucer, then...

FLYING—March 1967

the prowling mind of Henri Coanda

by G. Harry Stine

IN 1910, the first jet airplane was flown near Paris. In 1915, the first strategic bomber with a range of 1,000 miles was flown in France.

In 1918, a rocket cannon similar to the present "bazooka" antitank rocket was tested near Le Havre, France, several months before Dr. Robert H. Goddard built a similar device in America.

In 1921, prefabricated houses of steel and concrete were being erected in France and Belgium.

In 1935, the first flying saucer was designed.

The man responsible for all of these sweeping and advanced inventions is Dr. Henri Coanda, a Rumanian scientist and engineer who lives in Paris. With over 33 basic United States patents in his name covering a range of subjects from the first jet engine (1914) to soil regeneration and sea-water desalinization, Dr. Coanda certainly ranks as the most inventive man alive.

He has been inventing for over 50 years and is still at it. Unlike many other inventors, Dr. Coanda turns out new devices that are not merely improvements on established ways of doing things, but advanced new approaches to the problems of the modern world.

Take his jet airplane, for example. In 1906, while visiting the aeronautical pioneer Captain Ferdinand Ferber, Coanda saw Ferber's box-kite airplane powered by a propeller. He told Ferber, "It is the string pulling your kite that bothers me," and proceeded to reveal to Ferber the concept of jet propulsion of aircraft that had flashed into his mind at that instant. It took him four years and $200,000 to turn his concept into a reality.

No single man today could easily afford this sort of expenditure, but General Constantine Coanda of Rumania, his

father, provided most of the funds. To gain the necessary knowledge to build his jet, Henri Coanda attended the University of Liege, Belgium, and graduated in the first class of the *Ecole Superieur de Aeronautique*, the world's first aeronautical engineering school.

Coanda had to invent nearly everything for his jet airplane. He went to such aviation pioneers as Ernest Archdeacon, Gustave Eiffel, Paul Painleve and Louis Bleriot, but they could not help him. Coanda decided to experiment to get the information on airfoils, structures, materials and stability that he needed. He built a wind tunnel in 1909 that permitted him to photograph air flow around various shapes. To learn about supersonic airflow for the design of his jet engine, he developed the first method of photographing bullets in flight.

To gather aerodynamic data on airfoils, he got permission from the authorities to mount a test balance in front of a railway locomotive. During the winter nights of [illegible], Coanda rode on the front of the locomotive traveling at high speed between Paris and St. Quentin making his measurements with the night wind whipping past him. The result was the world's first thick airfoil section, quite a contrast to the thin cambered wings then used on airplanes.

His jet engine was unique. A 50 horsepower water-cooled piston engine built by Pierre Clerget, drove a centrifugal compressor that sucked in air and blasted it back into a shroud at high pressure. Coanda then ducted the Clerget engine exhaust into the compressed air, added a spray of raw fuel and ignited the mixture. The result was a [illegible] in England in 1939 generated just 600 pounds of thrust.)

The airplane in which Coanda mounted this engine was as unusual as the engine itself. It was constructed completely of molded mahogany plywood. Its two thick wings were cantilevered from simple struts at their midsections with none of the wires and braces common to airplanes at the time. The landing gear retracted into the lower wing. The gasoline tank was buried in the thick upper wing. The aircraft was aerodynamically clean, fast and far ahead of its time in 1910.

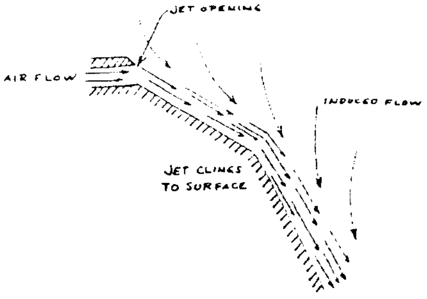
Henri Coanda tested his jet airplane at Issy-les-Moulineau outside Paris on December 10, 1910. Here is how he described the test of the first jet airplane:

"I had not intended to fly the plane that day, but merely to determine how to control it on the ground with its jet running. The engine worked well in the shop. But when I mounted it in the aircraft, the heat from the two jet exhausts coming back alongside the cockpit was too much for me. Therefore, I added two simple flat plates on the top and bottom of each jet exhaust to keep the heat away from me. The exhausts were, of course, angled away from the wooden fuselage so that they would not set the plane afire. When I got into the open cockpit and started moving the airplane along the ground, I saw at once that something was wrong. The jet flames would not stay in their asbestos exhaust pipes. They came out of the pipes and hugged the fuselage sides. Being afraid that the plane would catch fire, I gave my full attention to adjusting the jet engine and did not notice that the craft was gaining speed. Then I looked up. The walls of Paris were right in front of me! I had no room to stop or turn around, so I tried to fly over them. But there had been no one to teach me how to fly, and my machine was new and different. I made the machine climb too steeply, and it stalled. The left wing dropped, and my plane crashed to the ground. I was not wearing a seat belt in the open cockpit, so I was thrown clear of the crash. I came to, feeling as though every tooth in my head had been jarred loose. My jet airplane was burning a few yards away."

In spite of the crash, it was the first jet airplane. Major Victor Houart, an eyewitness, has stated that the craft did fly and described how cavalry officers exercising their mounts on the field at the time had to gallop out of the way of the plane as it flew at low altitude across the field before it climbed and stalled.

Henri Coanda was granted his first U.S. patent (No. 1,104,963) in 1914. It covered not only the jet engines, but also a fluid transmission for automobiles.

The Coanda Effect concerns the tendency of a moving jet of air or water to adhere to an adjacent surface. It has wide application in the growing new field of fluid valves and amplifiers.



BASIC COANDA EFFECT

A/Pilot's cabin
B/Moving platform
C/Fixed platform
D/Universal joint (entire cabin rotates to face direction of flight)
E/Gyroscopes
F/Coanda nozzles
G/Steam collector
H/Steam feed pipe
I/Horizontal blowing nozzle
J/Collector
K/Steam cooler & heat exchange
L/Boiler
M/Burners
N/Collector
O/Universal joint
P/Lifting devices (these tilt for control)
Q/Perforated top surface of wing
R/Lower surface of machine
S/Water tank
T/Fuel tank

Right: This VTOL machine has no moving parts, derives lift and forward thrust from Coanda Effect, and is steam-powered. The operating principle is similar to boundary layer control, and the jet flap.

A year later, Coanda designed and built the world's first twin-engined airplane, which he flew at the military competition at Reims, France. Again, his inventive genius created the new and unusual. The early rotary airplane engines were plagued with a great deal of torque because of the fact that the crankshaft was bolted to the plane and was therefore stationary while the cylinders and crankcase, attached directly to the propeller, spun around. The large mass of the rotating engine created not only a gyroscopic effect but also a torque that tended to cause an airplane to roll. To eliminate this problem, Coanda mounted two engines on opposite sides of the fuselage so that their torque cancelled out. Their power was transmitted through a differential gear box similar to that in the rear end of an automobile, and this in turn spun a single shaft that drove a four-bladed propeller. The machine flew well, but more conventional wire-braced box-kite types dominated the competition.

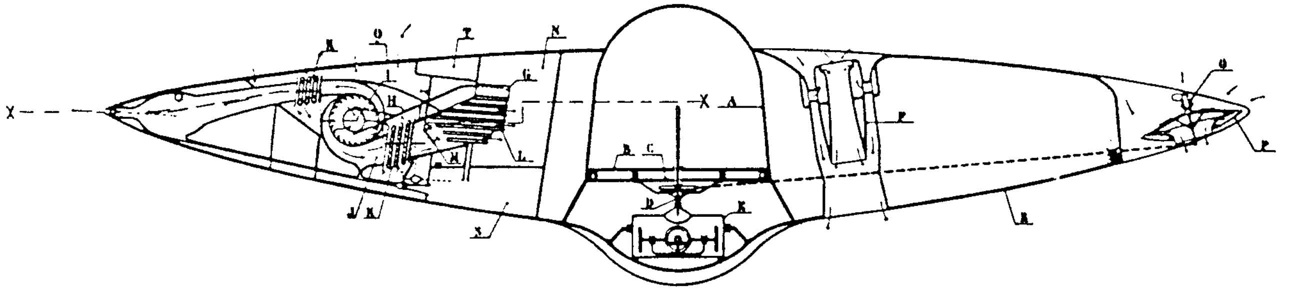
However, Coanda's activities came to the attention of Sir George White, who brought him to England to become chief engineer of the Bristol Aeroplane Company, Ltd. While at Bristol from 1912 to 1914, Coanda exerted a profound influence on the British aircraft industry with his Bristol-Coanda Military Monoplane, the most advanced aircraft in the world at the time.

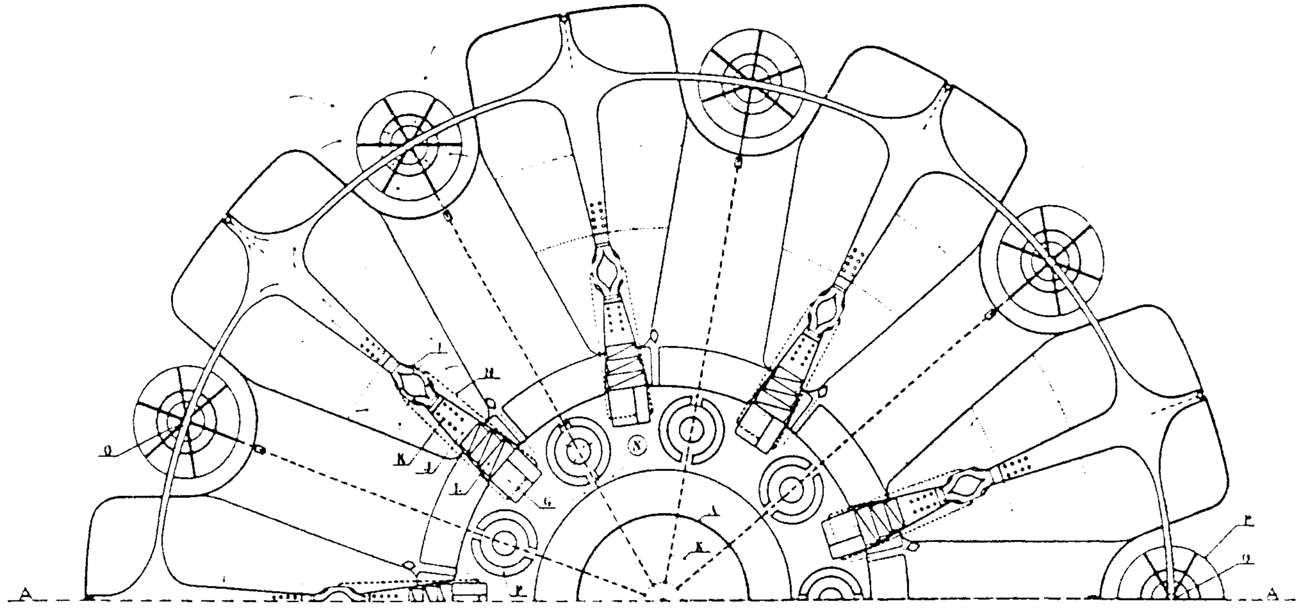
One of Coanda's most active periods was during World War I when he served the French government as an airplane builder. In 1916, his two-place strategic bomber covered a distance of 1,000 miles nonstop during tests. The war ended, however, before he got it into production.

In another military-inspired move, Captain Charles Nungesser, the famous French aviator, asked Coanda to develop a potent airborne attack weapon. The result was a 5.5-inch rocket-powered shell. Coanda developed the entire system—launching tube, rocket engine and shell. It was tested at Le Havre in August, 1918, and had a range of over a mile. Nungesser intended to mount this on fighter aircraft because it had no recoil. This pioneer work by Coanda was several months ahead of similar work in America by Dr. Robert H. Goddard.

In 1932, Coanda turned to research and invention as a full-time occupation.

Coanda's Lenticular Aerodyne
(Flying Saucer to you)





Within months, he had discovered the famous Coanda Effect, the tendency of a moving jet of air or water to adhere to and be deflected by a surface adjacent to it.

Coanda Effect, so named by Professor Albert Metral, who performed the mathematical analysis of it in 1939, is the key to the new field of "fluid amplifiers." In these devices, now emerging from the laboratories of several large companies, molecules of gas or liquid are used in ways analogous to the way electrons are used in high-fidelity amplifiers or radios. In electronics, the concept of electrical current is often described as similar to a flow of water, while electrical voltage is shown to be similar to water pressure. Fluid amplifiers have made this analogy a reality. In electronic amplifiers, electrons are used to control the flow of other electrons; in fluid amplifiers, thanks to Coanda Effect, air is used to control the flow of air. Fluid amplifiers can be designed to use any kind of fluid, including gases, water, oil or alcohol.

Think of any device that uses air or water, and the chances are that fluid amplifiers will soon replace electrical devices in them. The reasons for this are many. The fluid amplifier can be made smaller. It can operate over wide temperature ranges. It can operate for a long time because the only moving parts are molecules of liquid or gas, and it can operate at lower cost.

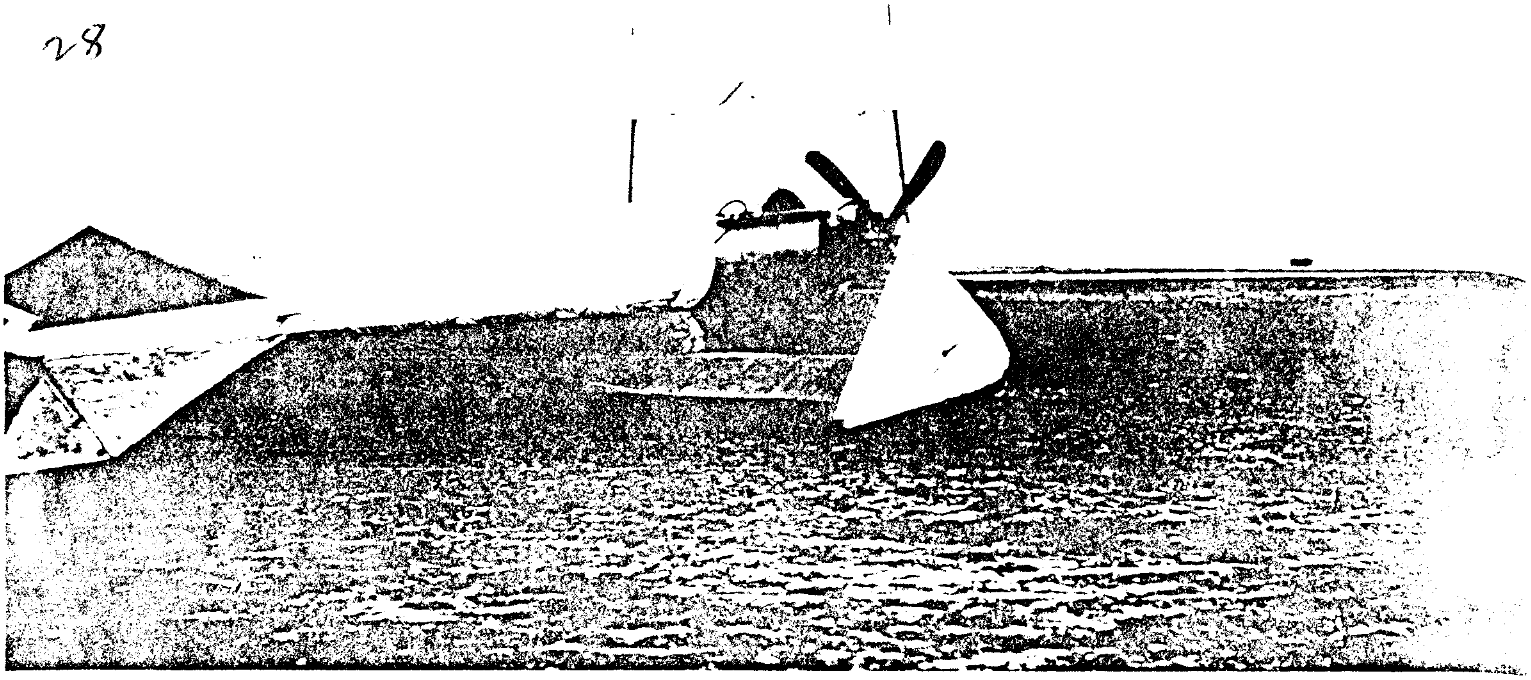
Henri Coanda has spent nearly 35 years applying his Effect to such devices as agricultural sprayers, fog generators, pumps, blowers, engine silencers, separators, marine propulsion systems, atomizers, carburetors, jet thrust reversers, high-lift devices for aircraft, jet engines and vertical takeoff and landing (VTOL) aircraft. He holds U.S. and foreign patents on all of these applications.

In 1935, he designed the first flying saucer. Coanda Effect can be used to create vertical lift without forward motion. He not only used Coanda Effect to provide lift, but also to control the craft. The Coanda Lenticular Aerodyne has never been built because Coanda has never had the money to do it. However, it is a feasible device. Actual tests on the critical portions of it have been made in France, and the results have indicated that it is practical.

However, Henri Coanda has not restricted his inventive abilities to mechanical gadgets alone. Back in 1914, he became interested in biology, and his ever-inquiring mind led him to conduct some of the first experiments in biological engineering. While measuring the build-up of static electrical charges on aircraft in flight (the static discharge "tails" on the wings of jet aircraft are there because of Henri Coanda), he also discovered that plants have an electrical charge opposite to the ground on which they are growing. Later, he measured the "heartbeat" of plants, the waves of pressure that sends sap upward in the stems and trunks of growing things. To improve the yield of his chateau gardens near Auxance, he built an "artificial cow" that produced fertilizer free of toxins and ready for immediate use. Then he developed a truly organic fertilizer, Bio Edophos.

Coanda has grown tobacco in a district of France where it has never grown before. He has produced plants with gigantic flowers and fruits, all without hybridization. He is one of the first biological engineers.

In 1945 Coanda began to study water problems as civilization's most basic



...a's ...a, claimed the world's first, had two engines driving a single, four-bladed propeller. Reims, France, 1911.

terest. He soon discovered crystalline water in living cells that was ice at 98 degrees F. and which melted at about 116 degrees F. He studied snowflakes and identified 10 different types or "molecular isotopes" of water in snow crystals alone. Soon, he was building devices for desalting sea water.

The prowling mind of Henri Coanda has recently become interested in the problems of handicapped people. He feels it may be possible to restore sight to the blind or hearing to the deaf by utilizing other nerve pathways to the brain.

At the age of 80, Dr. Henri Coanda is still inventing and is a healthy and active man. His trained and experienced mind grasps problems in a unique way. He anticipates the needs of society. He comes up with unexpected answers because he tackles a problem with no preconceived notions of how to solve it. The inevitable question arises: why have we not heard of him? In this age of rapid progress in technology, why hasn't his work been snapped up at once? The answer is simple: Henri Coanda prefers to work alone.

In today's world of research and development teams, who will listen to a lone inventor?

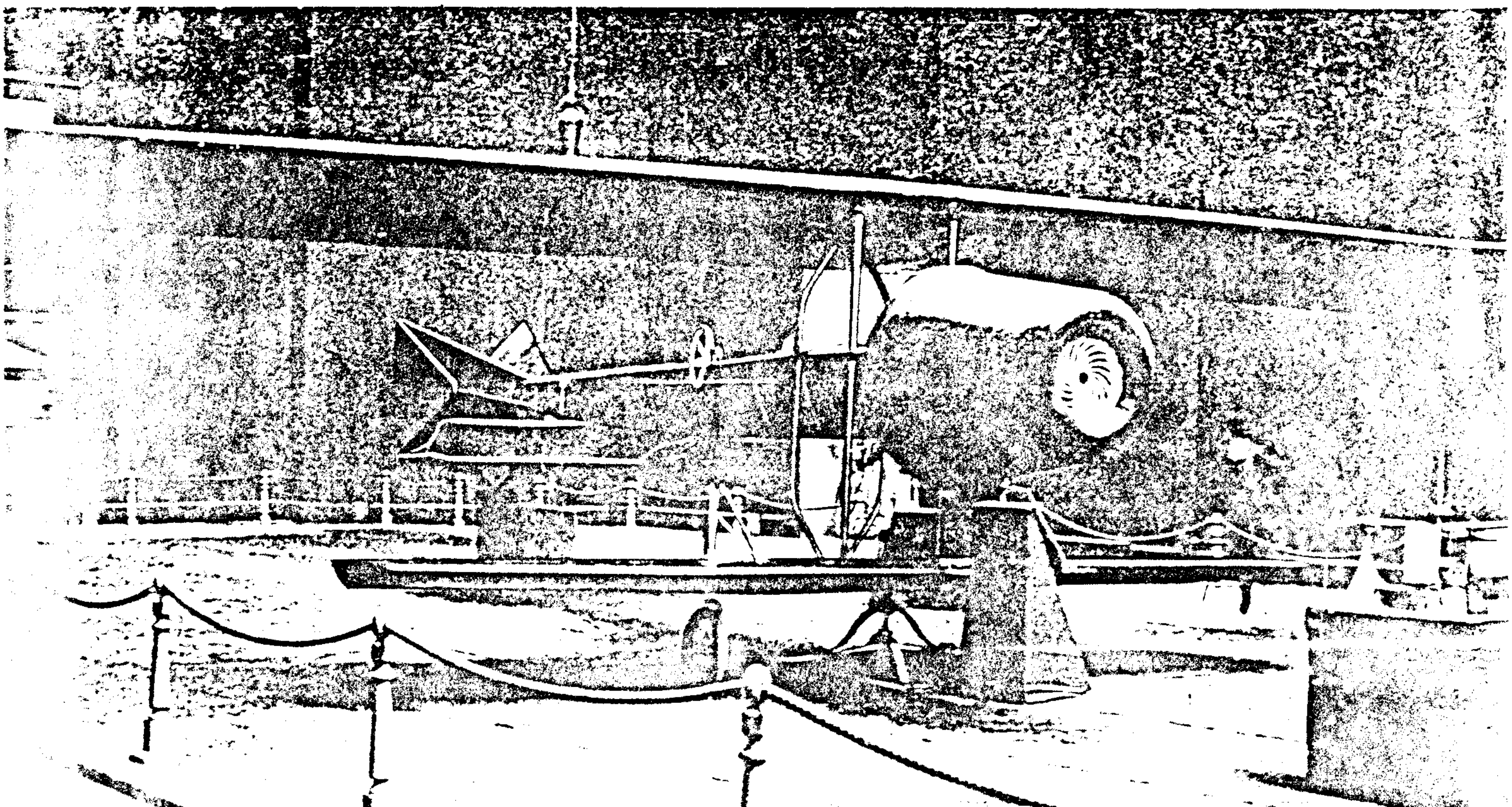
People are listening to him now.

"It is the means which create the need," he states. "Before the telephone was invented, nobody needed it. Now, nobody can do without it. Before the automobile was invented, nobody wanted it; now, nobody can do without it."

We'll be hearing more in the future from the most inventive man alive. ✝

The author has an extensive background in science and aerospace technology. As an assistant director of research for a U.S. firm, he worked as a colleague with Dr. Coanda for several years in the 1960s.

The Coanda jet put out 484 pounds of thrust. It crashed on its maiden flight, from Coanda's inexperience at piloting.



Henri Coanda--U.S. Patents

(*complete patent included
in this volume.)

2,052,869
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2,187,342
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Feb. 15, 1938. H. COANDA 2,108,652

PROPELLING DEVICE

Filed Jan. 10, 1936 2 Sheets-Sheet 1

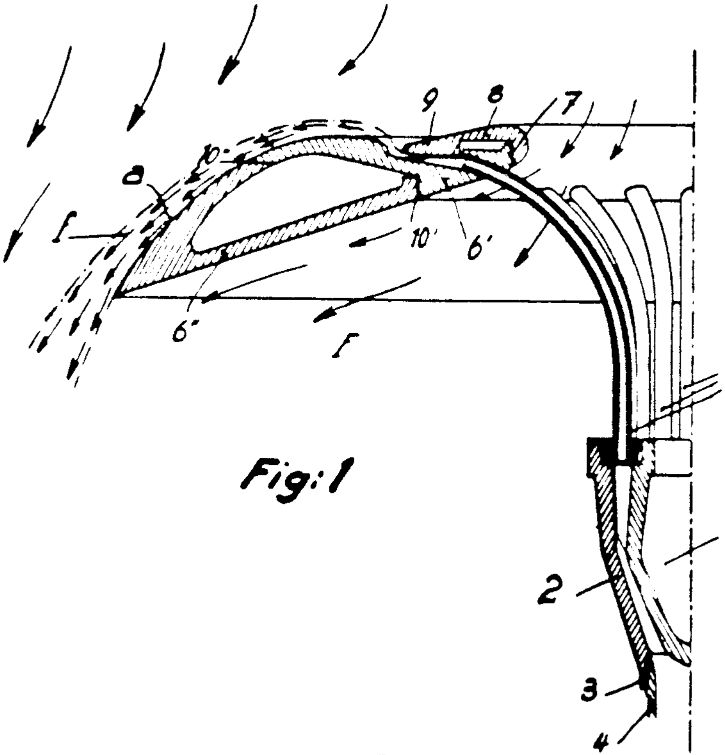

Fig: 1

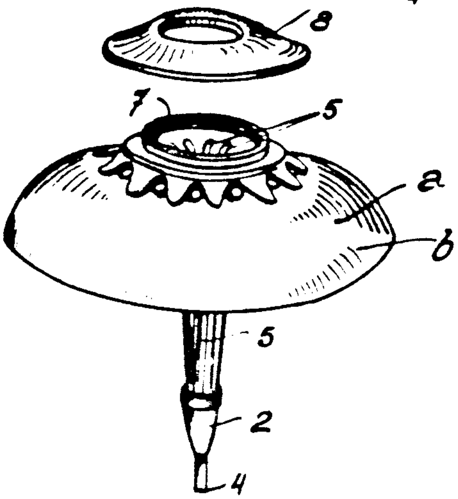

Fig: 2

INVENTOR.
HENRI COANDA
BY Haseltine, Lake & Co
ATTORNEYS

Feb. 15, 1938. H. COANDA 2,108,652

PROPELLING DEVICE

Filed Jan. 10, 1936 2 Sheets-Sheet 2

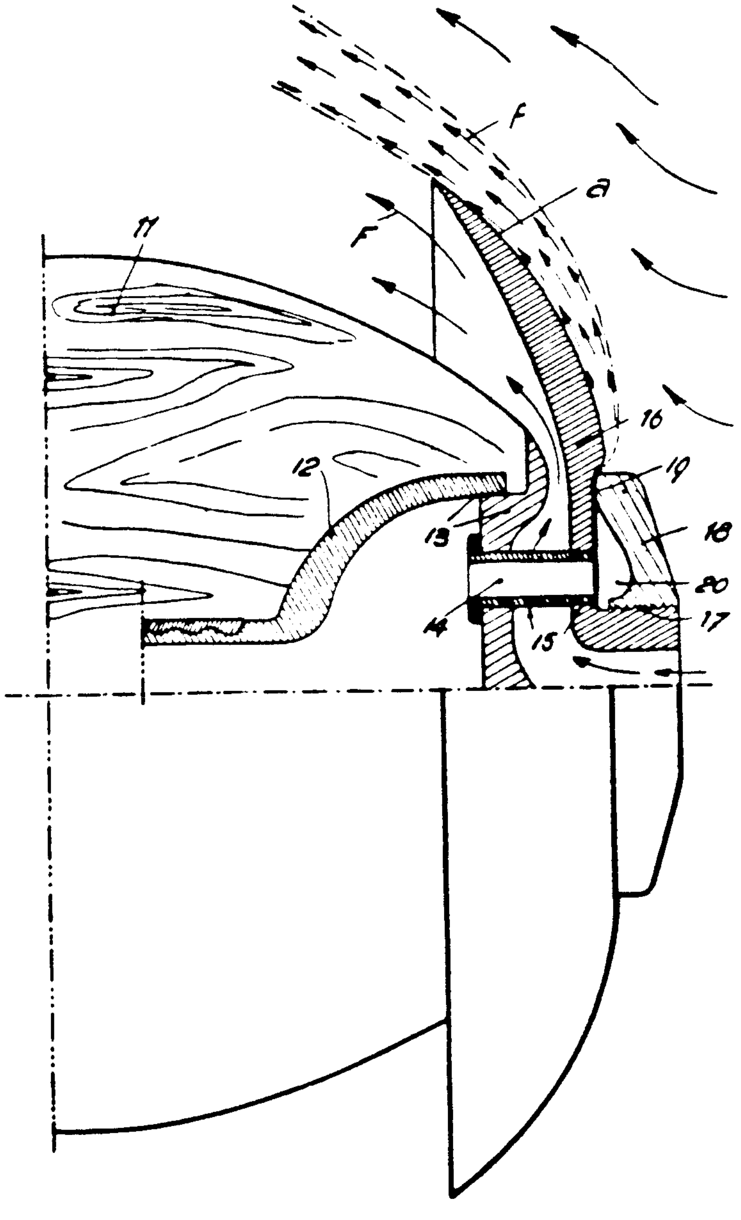

Fig: 3

INVENTOR:
HENRI COANDA
BY Haseltine Lake & Co
ATTORNEYS

Patented Feb. 15, 1938 2,108,652

UNITED STATES PATENT OFFICE

2,108,652

PROPELLING DEVICE

Henri Coanda, Clichy, France, assignor to Société Anonyme d'Etudes des Brevets et Procédés Coanda-Société Coanda, a corporation of France

Application January 10, 1936, Serial No. 58,471
In France January 15, 1935

10 Claims. (Cl. 244—73)

The present invention relates to propelling devices in which there is produced a suction zone in front of the body in motion on which the propeller is mounted, this suction being such that the body in motion is propelled under the influence of the atmospheric pressure existing at the rear of the propeller.

The object of the present invention is to provide a propelling device of this kind which is better adapted to meet the requirements of practice than those made up to the present time. According to the invention the propeller is composed of an annular structure of suitable axial section, on which a ring fits in an adjustable manner, said ring being concentric and constructed in such a manner that a very narrow gap is formed between it and said annular structure, through which gap compressed gas escapes outwardly along a frontwardly directed line, the axial section of said annular structure consisting of a curve tangent to this line and having its convexity toward the front, while the axial section of said ring is a line making a substantial angle with said first mentioned line.

Furthermore the ring and annular structure are hollow, at least to a certan depth so as to allow air from the space ahead of the propeller to flow through their central portion and out behind the annular structure. This prevents any suction at the rear of the annular structure so that the whole of the suction is in front thereof, and the motion of the body is ensured by the difference between the pressure at the front and that at the rear

Preferred embodiments of the present invention will be hereinafter described with reference to the accompanying drawings, given merely by way of example and in which

Figs. 1 and 2, show in perspective and sectional views respectively a first embodiment of the invention

Fig. 3 is a sectional view of a modification,

In the embodiment of Figs. 1 and 2, the body of the propelling device according to the invention includes an inner element **1** and a tubular part **2**, fitted thereon and provided at **3** with screw threads for fixation of a tube **4**. This structure carries a series of tubes **5** opening into an annular member **6**. Another annular member **8** is screwed at **7** on member **6**, so that the narrow annular interval **9** between members **6** and **8** can be adjusted at will.

Member **6** may be made of a single piece or consist of two rings **6'**, **6''** screwed to each other at **10'**, **10''**.

In the embodiment of Fig. 3, the streamline body **11** contains a part screwed to the member **13**. Tubes **14** are provided in member **13** and secured by the screw thread **15** in a ring **16** of suitable profile upon which a ring **18** is fixed by screw threads **17**, so that this ring **18** can be adjusted in position with respect to ring **16**.

Between ring **16** and ring **18** there is left an adjustable narrow annular interval **19**, ring **18** being however constructed so as to form a circular chamber **20** between said ring and ring **18**.

The propeller works in the following manner:

The compressed gas, which may consist of superheated steam or a combustible mixture or some explosive mixture or even compressed air, is supplied through tube **4**, flows through the annular space formed between elements **1** and **2**, passes through tubes **5** into the narrow annular interval **9** between the element **6** and ring **8** and escapes into the atmosphere.

The fluid film expands and exceeds its initial volume, and, owing to the fact that on one side of the outlet of passage **9**, annular member **6** is substantially tangent to the fluid sheet escaping from **9**, whereas, on the other side, the edge of member **8** makes a substantial angle to the direction of said sheet, the latter flows along the front face of member **6**, following the path indicated by arrows *f*. Thus in the space marked in dotted lines a rush of fluid is created toward which the surrounding air is drawn in the direction of the arrows *f* provided the pressure of the motive fluid is sufficient.

Therefore the propeller, owing to the suction created at its front part, has a tendency to rush frontward under the action of the pressure existing behind it, drawing along the structure on which it is mounted provided the outflow is sufficient to create a momentum at low speed and in great masses of the surrounding air, equivalent to the drag of the body in which the propeller is mounted.

Moreover the surrounding air circulates through the central free portion in the direction of the arrows **F**, at same pressure, so that below the part **6** a counterpressure is created relative to the suction in front of the same, which counter pressure consequently adds to the action of the suction created in front of part *a*.

The same applies in the case of Fig. 3. The gaseous mixture under high pressure which is to expand after entering the chamber formed by part **12**, passes through tube **14**, starts expanding in chamber **20** and escapes through annular interval **19** following the front face of element **16**, creating in front of it a zone of very strong suction represented by dotted lines.

It is evident that the shape of elements such as 6' and 16 and the thickness of annular intervals 8 or 18 are calculated in such a manner that a certain expansion of the compressed motive fluid permits of obtaining a momentum which ensures the displacement of the body.

In a general manner, while I have, in the above description, disclosed what I deem to be practical and efficient embodiments of the present invention, it should be well understood that I do not wish to be limited thereto as there might be changes made in the arrangement, disposition and form of the parts without departing from the principle of the present invention as comprehended within the scope of the appended claims.

It may be noted that in Fig. 1 the member 8 and in Fig. 3 the part 18 extend at least with the edge into a recess at the front of the moving body, with the result that the air or fluid issuing from the narrow slot first strikes the curved portion of the airfoil or wing shaped member so as to follow its curvature forwardly and then outward radially and subsequently rearwardly.

Having now fully described my invention, I claim:

1. A propelling device of the type described, which comprises, in combination, a body, an annular member rigid with said body at the front part thereof, another annular member rigid with said body, coaxial with the first annular member and located ahead thereof, so as to leave between said members a narrow annular interval with a flaring outlet opening to the atmosphere in a frontward direction, the rear edge or lip of said outlet, formed by the front surface of the first mentioned annular member, being of rounded shape starting tangentially to said frontward direction on the outer side thereof, with its convexity toward the front, whereas the front edge or lip of said outlet, formed by the front surface of the second mentioned annular member, starts at an angle to said direction on the inner side thereof, and means for driving out a fluid under high pressure through said annular outlet.

2. A propelling device of the type described, which comprises, in combination, a body, an annular member rigidly fixed to said body at the front part thereof, another annular member rigidly fixed to said body in coaxial relation with the first annular member and ahead thereof, said members being so positioned and shaped as to form between them a narrow annular interval with a flaring outlet opening to the atmosphere in a frontward direction, the rear edge or lip of said outlet, formed by the front surface of the first mentioned annular member, being of rounded shape starting tangentially to said frontward direction on the outer side thereof, with its convexity toward the front, whereas the front edge or lip of said outlet, formed by the front surface of the second mentioned annular member, starts at an angle to said direction on the inner side thereof, and means for driving out a fluid under high pressure through said annular outlet.

3. A propelling device of the type described, which comprises, in combination, a body, an annular member rigidly fixed to said body at the front part thereof, with a space between the rear face of said member and said body, whereby air from the front of said member can flow through the inner aperture thereof and through said space to the lateral sides of said body, another annular member rigidly fixed to said body in coaxial relation with the first mentioned annular member and ahead thereof, said members being so positioned and shaped as to form between them an annular interval with a flaring outlet opening to the atmosphere in a frontward direction, the rear edge or lip of said outlet, formed by the front surface of the first mentioned annular member, being of rounded shape starting tangentially to said frontward direction on the outer side thereof, with its convexity toward the front, whereas the front edge or lip of said outlet, formed by the front surface of the second mentioned annular member, starts at an angle to said direction on the inner side thereof, and means for driving out a fluid under high pressure through said annular outlet.

4. A propelling device of the type described, which comprises, in combination, a body, an annular member rigidly fixed to said body at the front part thereof, with a space between the rear face of said member and said body, whereby air from the front of said member can flow through the inner aperture thereof and through said space to the lateral sides of said body, another annular member rigidly fixed to said body, in coaxial relation with the first mentioned annular member, and ahead thereof, said members being so positioned and shaped as to form between them an annular chamber having an annular flaring outlet opening to the atmosphere in a frontward direction, the rear edge or lip of said outlet, formed by the front surface of the first mentioned annular member, being of rounded shape starting tangentially to said frontward direction on the outer side thereof, with its convexity toward the front, whereas the front edge or lip of said outlet, formed by the front surface of the second mentioned annular member, starts at an angle to said direction on the inner side thereof, and means for feeding a fluid under high pressure to said annular chamber between said members, the last mentioned means including a plurality of tubes extending from said body and opening into said chamber.

5. A propelling device of the type described which comprises, in combination, a body of hollow shape forming a container for a fluid under high pressure, an annular member rigidly fixed to said body at the front part thereof, with a space between said body and the rear face of said member, whereby air can flow from the front of said annular member, through the central aperture thereof and through said space, to the lateral sides of said body, another annular member rigidly fixed to said first mentioned annular member coaxially therewith and ahead thereof, said members being so positioned and shaped as to form between them an annular chamber having an annular flaring outlet opening to the atmosphere in a frontward direction, the rear edge or lip of said outlet, formed by the front surface of the first mentioned annular member, being of rounded shape starting tangentially to said frontward direction on the outer side thereof, with its convexity toward the front, whereas the front edge or lip of said outlet, formed by the front surface of the second mentioned annular member, starts at an angle to said direction on the inner side thereof, and a plurality of pipes connecting the inside of said body with said annular chamber for feeding fluid under high pressure to said chamber.

6. A propelling device of the type described, which comprises, in combination, an elongated body having one end serving as a forward portion or nose, a fixed surface member of wing-shaped section rigidly carried exteriorly upon the forward portion of said body in spaced relation to the latter and with the chord of said section practically transversely disposed with respect to the forward direction upon the body, the front surface of said member being convex in said forward direction and arching rearwardly away from said body from a recess or depressed portion in the foremost part of said forward portion, rigid spaced means upon said fixed member spaced a small distance from the latter member in said recess or depressed portion so as to determine at least one narrow slot communicating with the interior of said body and allowing a sheet of fluid to be projected at high pressure from within the body out through said slot and against the convex front surface of said fixed member in said recess and caused to follow said curved surface first forwardly out of the recess and then outwardly and rearwardly along the arching portion of said curved surface upon said fixed member, and open means upon the forward portion of said propelling device allowing free access of air to the rear surface of said fixed surface member between the same and the body independently of the fluid supply escaping from the interior of said body through said narrow slot.

7. A propelling device according to claim 6, wherein the fixed surface member is annular so as to surround the nose of the device, and the slot means extends around said nose in the recess which is also extended about the device to be annular in form.

8. A propelling device of the type described, which comprises, in combination, an annular elongated body having one end serving as a forward portion or nose, a fixed annular surface member of wing-shaped section rigidly carried exteriorly upon the forward portion of said body in spaced relation to the latter and surrounding the same with the chord of said section practically transversely disposed with respect to the forward direction upon the body, the front surface of said annular member being convex in said forward direction and arching rearwardly away from said body, rigid spaced means upon said fixed member spaced a small distance from the latter member so as to determine at least one narrow slot communicating with the interior of said body and allowing a sheet of fluid to be projected at high pressure from within the body out through said slot and caused to follow said curved surface outwardly and rearwardly along the arching portion of said curved surface upon said fixed and annular member, there being a free and open passage extending rearwardly from the forward portion of the device and communicating in unbroken manner with the space between said fixed annular member and said body allowing atmospheric air to enter freely and relieve any tendency to form a vacuum upon the rear surface of said annular member, and a plurality of fluid supply tubes connecting the interior of said body with said narrow slot through said space between the annular fixed member and said body and independently thereof.

9. A propelling device according to claim 8, wherein the fluid supply tubes serve as the exclusive mechanical means for supporting the fixed annular member upon and spacing the same away from the main body of the device.

10. A propelling device of the type described, which comprises, in combination, an elongated body having one end serving as a forward portion or nose, a fixed annular surface member of wing-shaped section rigidly carried exteriorly upon the forward portion of said body in spaced relation to the latter and surrounding the same with the chord of said section practically transversely disposed with respect to the forward direction upon the body, the front surface of said annular fixed member being convex in said forward direction and arching rearwardly away from said body from an annular recess or depressed portion in the foremost part of said forward portion, rigid spaced means upon said fixed member spaced a small distance from the latter member in said recess or depressed portion so as to determine at least one narrow slot communicating with the interior of said body and allowing a sheet of fluid to be projected at high pressure from within the body out through said slot and against the convex front surface of said fixed member in said annular recess and caused to follow said curved surface first forwardly out of the recess and then outwardly and rearwardly along the arching portion of said curved surface upon said fixed annular member, there being a free and open passage extending rearwardly from the forward portion of the device and communicating in unbroken manner with the space between said fixed annular member and said body allowing atmospheric air to enter freely and relieve any tendency to form a vacuum upon the rear surface of said annular member, and a plurality of fluid supply tubes connecting the interior of said body with said narrow slot through said space between the annular fixed member and said body and independently thereof.

HENRI COANDA.

May 9, 1939. **H COANDA** 2,157,281

PROPELLING DEVICE

Original Filed Jan. 10, 1936 2 Sheets-Sheet 1

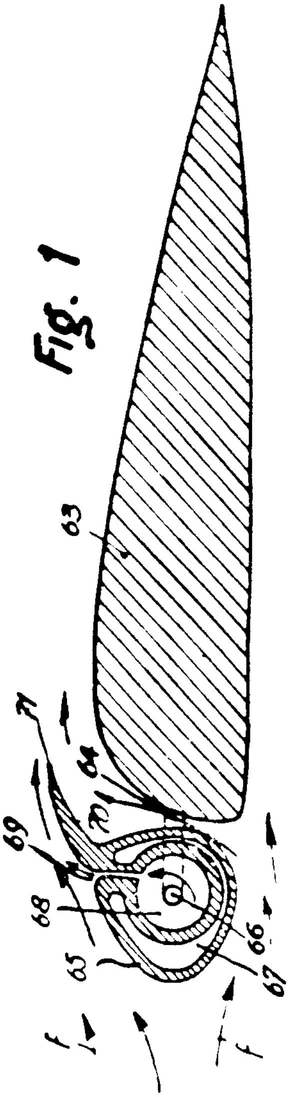

Fig. 1

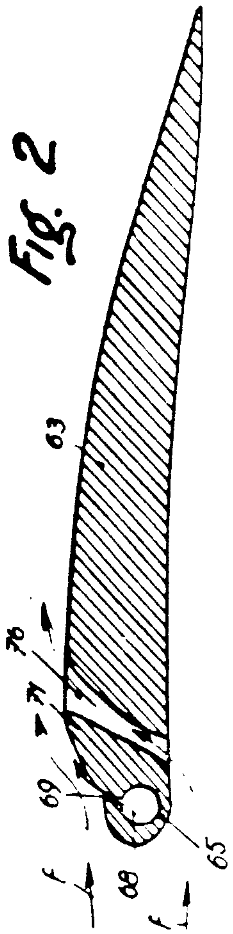

Fig. 2

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May 9, 1939. H COANDA 2,157,281

PROPELLING DEVICE

Original Filed Jan. 10, 1936 2 Sheets-Sheet 2

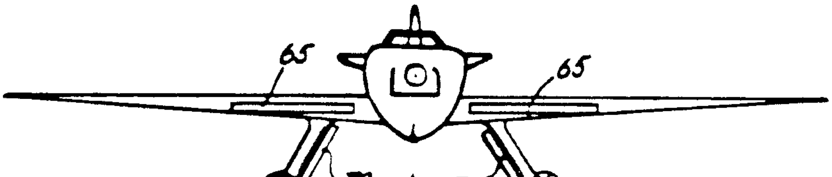

Fig: 3

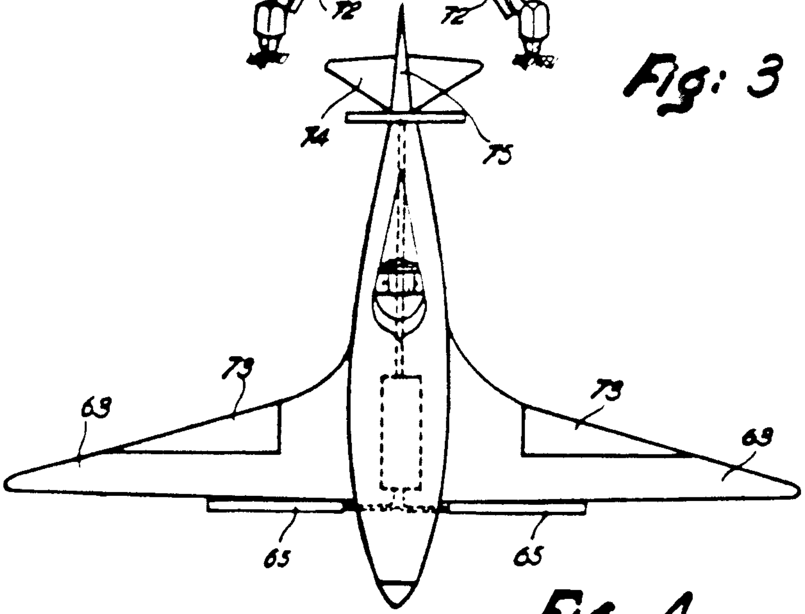

Fig. 4

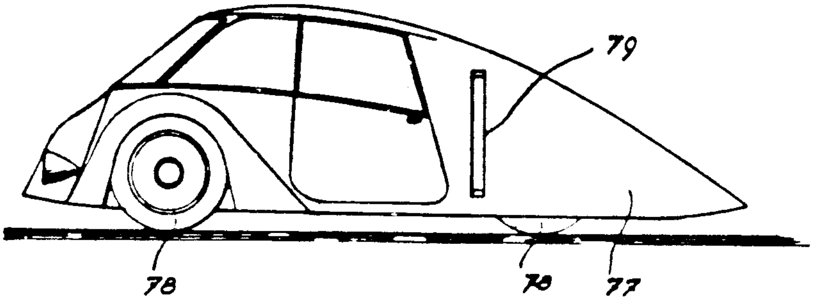

Fig. 5

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Patented May 9, 1939 **2,157,281**

UNITED STATES PATENT OFFICE

2,157,281

PROPELLING DEVICE

Henri Coanda, Clichy, France, assignor to Société Anonyme d'Etudes des Brevets et Procédés Coanada-Société Coanda, a corporation of France

Original application January 10, 1936, Serial No. 58,471, now Patent No. 2,108,652, dated February 15, 1938. Divided and this application August 13, 1936, Serial No. 95,818. In France January 16, 1935

8 Claims. (Cl. 244—15)

This application is a division of my U. S. patent application Ser. No. 58,471, filed January 10, 1936, now Patent Number 2,108,652 issued Feb. 15, 1938.

The invention relates to the propulsion of vehicles by the direct action of a fluid under pressure escaping into the atmosphere.

The object of the invention is to provide a propelling system which is better adapted to meet the requirements of practice than those applied up to the present time.

The essential feature of the present invention consists in the provision of a cylinder adapted to be fitted along a surface of the vehicle, preferably at the front part thereof, said cylinder being hollow and fed with a fluid under pressure, and communicating with the atmosphere through an outlet narrow slot extending longitudinally, the rear edge or lip of said slot being prolonged by a convex rounded part tangent to the outlet direction of said slot, whereas the front lip makes directly a substantial angle with said direction.

Other features of the present invention will result from the following detailed description of some specific embodiments thereof.

Preferred embodiments of the present invention will be hereinafter described with reference to the accompanying drawings, given merely by way of example, and in which:

Fig. 1 shows the propelling system according to the present invention in transverse section as applied to the wing of an airplane;

Fig. 2 is a similar sectional view of a modification;

Figs. 3 and 4 show airplanes in front and top plan views, respectively, as fitted with the propelling device according to the invention;

Fig. 5 is a side elevation of an automobile vehicle fitted with the propelling device according to the present invention.

In the embodiments of Figs. 1 and 2, the propeller 65 is fixed at the front of a wing 63, to which it is connected through any suitable means, such as 64. This propeller is adapted to rotate about axis 66. It consists of a cylinder 67, preferably hollow, so as to reduce its weight, and provided with an inner chamber 68 filled with gas under pressure. This chamber 68 communicates with the atmosphere through a narrow slot 69. The rear edge or lip of the outlet of said slot is prolonged by a convex rounded surface 70 terminating in a point 71. On the contrary the front lip of said outlet directly makes a substantial angle with the outlet direction.

With such an arrangement, the fluid escaping through slot 69 is caused to flow along surface 70 and it creates a suction ahead of the propeller which, if a sufficient amount of gas under pressure is caused to escape at a pressure sufficiently high, can exert a driving action on the airplane wing and propel it at the desired speed.

Furthermore, a portion of the air surrounding the system is caused to flow in the direction of arrows *f*, a portion of this air passing under cylinder 67 and between said cylinder and the front edge of the wing, which creates a counter pressure under point 71, the action of said counter-pressure being added to that of the suction created ahead of said point.

If, through any suitable means, cylinder 67 is caused to turn about its axis 66, the direction of the suction acting upon part 70—71 can be modified. In particular, for taking off, it will be advantageous to turn part 70 in an anti-clockwise direction from the position shown by Fig. 1, in such manner that said part 70 makes with respect to the ground a much greater angle than when the airplane is flying a horizontal course.

In the embodiment of Fig. 2, chamber 68 is formed directly in the wing. In this position of member 70, the suction will be directed in an upward direction, thus increasing the lift. The paths of the fluid and air are substantially the same as in the preceding embodiment.

Figs. 3 and 4 show two different applications of this propelling device to an airplane. In both cases, propellers 65 are fixed to the leading edge of the wings. But in the embodiment of Fig. 3, similar propellers are also fixed to the uprights of the landing gear, at 72. The control of the airplane is ensured by means of the ailerons and rudders 73, 74, 75, operated in the usual manner.

Fig. 5 shows a motor car body 77, of streamline shape, with fairings for the wheels 78.

In this embodiment of the invention, a propeller of the kind of that shown by Fig. 1 is fitted vertically on the outer wall of the body, for instance at 79, or at the front.

In a general manner, while I have, in the above description, disclosed what I deem to be practical and efficient embodiments of the present invention, it should be well understood that I do not wish to be limited thereto as there might be changes made in the arrangement, disposition and form of the parts without departing from the principle of the present invention as comprehended within the scope of the accompanying claims.

What I claim is:

1. In a body member adapted to be propelled in a given forward direction having an elongated

hollow leading edge transversely disposed with respect to said direction, and there being two spaced slots running within said leading edge in general parallelism therewith, the combination of features consisting in that the hollow leading edge is adapted interiorly to be fed with a fluid under pressure which issues through the foremost slot and is of rounded exterior section of less size than the general section of the body member but practically tangent with the lower surface thereof, in that said first slot is narrow, unalterable, communicates upwardly in a direction directly transverse to said given forward direction and terminates within the hollow interior of the leading edge, in that the second slot passes entirely through said body member in an upward and unalterably rearwardly inclined manner, and in that the rear lip of the first slot between both slots curves upwardly and rearwardly in rigidly unalterable manner from the opening of said first slot so as to arch rearwardly to at least the full section of said body member.

2. In a body member adapted to be propelled in a given forward direction and having an elongated hollow leading edge transversely disposed with respect to said direction, and there being two spaced slots running within said leading edge in general parallelism therewith, the combination of features consisting in that the hollow leading edge is adapted interiorly to be fed with a fluid under pressure which issues through the foremost slot and is of rounded exterior section of less size than the general section of the body member but extends a distance downwardly toward the lower surface thereof, in that said first slot is narrow, unalterable, communicates upwardly in a direction generally transverse to said given forward direction and terminates within the hollow interior of the leading edge, in that the second slot passes entirely through said body member in an upward and unalterably rearwardly inclined manner, and in that the rear lip of the first slot between both slots curves upwardly and rearwardly from the opening of said first slot so as to arch rearwardly to at least the full section of said body member.

3. In connection with a body member adapted to be propelled in a given forward direction and having an elongated part transversely disposed with respect to said direction, a propeller which comprises, in combination, a hollow cylinder carried by said body in parallel spaced relation to said elongated part, said cylinder being adapted to be fed interiorly with a fluid under pressure and being formed, for the purpose of allowing said fluid to issue to the atmosphere, with an unalterable narrow outled slot extending along a generatrix of said cylinder and located in a plane transverse to said forward direction, a short front lip for said slot and a relatively longer rear lip for said slot which is rigid with said cylinder and has a rounded shape starting tangentially to said transverse plane with its convexity turned toward the front and curving gradually so as ultimately to become substantially tangential to said direction, the front lip of said outlet slot directly making an angle with said transverse plane, and said cylinder being located at a distance from the elongated part of said body so as to leave between said cylinder and said body a free space for the flow of air from the front of said cylinder to the space at the rear or underside of said lip.

4. In connection with a body member adapted to be propelled in a given forward direction and having a front or leading edge transversely disposed with respect to said direction, a propeller which comprises, in combination, a hollow cylinder carried by said body in parallel spaced relation to said front or leading edge, said cylinder being adapted to be fed interiorly with a fluid under pressure and being formed, for the purpose of allowing said fluid to issue to the atmosphere, with an unalterable narrow outlet slot extending along a generatrix of said cylinder and located in a plane transverse to said forward direction, a short front lip for said slot and a relatively longer rear lip for said slot which is rigid with said cylinder and has a rounded shape starting tangentially to said transverse plane with its convexity turned toward the front and curving gradually so as ultimately to become substantially tangential to said direction, the front lip of said outlet slot directly making an angle with said transverse plane, and said cylinder being positioned at a distance from the front or leading edge of said body so as to leave between said cylinder and said body a free space for the flow of air from the front of said cylinder to the space at the rear or underside of said lip.

5. In connection with a body member adapted to be propelled in a given forward direction and having a front or leading edge transversely disposed with respect to said direction, a propeller which comprises, in combination, a hollow cylinder carried by said body in advance thereof and in parallel spaced relation to said front or leading edge, said cylinder being adapted to be fed interiorly with a fluid under pressure and being formed, for the purpose of allowing said fluid to issue to the atmosphere, with an unalterable narrow outlet slot extending along a generatrix of said cylinder and located in a plane transverse to said forward direction, a short front lip for said slot and a relatively longer rear lip for said slot which is rigid with said cylinder and has a rounded shape starting tangentially to said transverse plane with its convexity turned toward the front and curving gradually so as ultimately to become substantially tangential to said direction, the front lip of said outlet slot turning sharply at about a right angle to said transverse plane toward the front, and said cylinder being positioned at a distance from the front or leading edge of said body so as to leave between said cylinder and said body a free space for the flow of air from the front of said cylinder to the space at the rear or underside of said lip.

6. In connection with a body member adapted to be propelled in a given forward direction and having a front or leading edge transversely disposed with respect to said direction, a propeller which comprises, in combination, a hollow cylinder carried by said body in advance thereof and in parallel spaced relation to said front or leading edge, said cylinder being adapted to be fed interiorly with a fluid under pressure and being formed, for the purpose of allowing said fluid issue to the atmosphere, with an unalterable narrow outlet slot extending along a generatrix of said cylinder and located in a plane transverse to said forward direction, a short front lip for said slot and a relatively longer rear lip for said slot which is rigid with said cylinder and has a rounded shape starting tangentially to said transverse plane with its convexity turned toward the front and curving gradually so as ultimately to become substantially tangential to said direction, the front lip of said outlet slot forming a sharp angled edge, said cylinder being positioned

at a distance from the front or leading edge of said body so as to leave between said cylinder and said body a free space for the flow of air from the front of said cylinder to the space at the rear or underside of said lip.

7. In connection with a body member adapted to be propelled in a given forward direction and having a front or leading edge transversely disposed with respect to said direction, a propeller which comprises, in combination, a hollow cylinder adjustably carried by said body member so as to be capable of being turned about an axis parallel to said front or leading edge and disposed in parallelism with said edge, said cylinder being adapted to be fed interiorly with a fluid under pressure and being formed, for the purpose of allowing said fluid to issue to the atmosphere, with an unalterable narrow outlet slot extending along a generatrix of said cylinder and located in a plane transverse to said forward direction, a short front lip for said slot and a relatively longer rear lip for said slot which is rigid with said cylinder and has a rounded shape starting tangentially to said transverse plane with its convexity turned toward the front and curving gradually so as ultimately to become substantially tangential to said direction, the front lip of said outlet slot forming a sharp angled edge, said cylinder being positioned at a distance from the front or leading edge of said body so as to leave between said cylinder and said body a free space for the flow of air from the front of said cylinder to the space at the rear or underside of said lip.

8. A self-propelled wing which comprises, in combination, a hollow front element forming the leading edge and adapted to be fed interiorly with a gas under pressure, and a rear element carrying the trailing edge of said wing, with a space between these two elements forming a slot extending toward the rear from the underside of the wing to the upper side thereof, said front element being formed for the purpose of allowing said gas to issue to the atmosphere, with an unalterable narrow outlet slot extending along at least a portion of said leading edge and located in a plane transverse to the fore and aft direction of the wing and substantially at right angles to the chord thereof, a short front lip and a relatively longer rear lip for said outlet slot, said rear lip of said slot starting tangentially to said transverse plane and curving gradually backward so as to form a prolongation of the rear upper face portion of said front hollow element, and said front lip of said outlet slot forming a sharp angled edge.

HENRI COANDA.

June 7, 1960 H. COANDA **2,939,654**

DEVICE FOR THE SIMULTANEOUS CONTROL OF
LIFTING AND DIRECTIONAL ELEMENTS

Filed April 30, 1957 3 Sheets-Sheet 1

Fig. 2

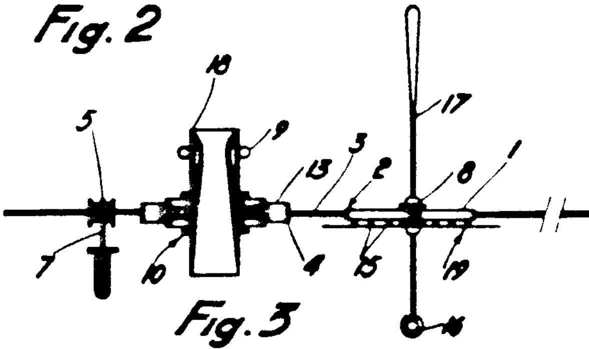


Fig. 4

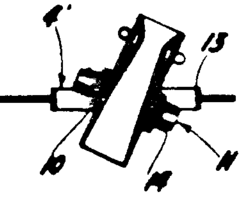


Fig. 3

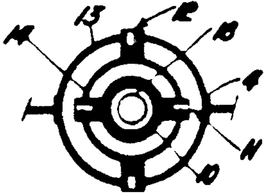
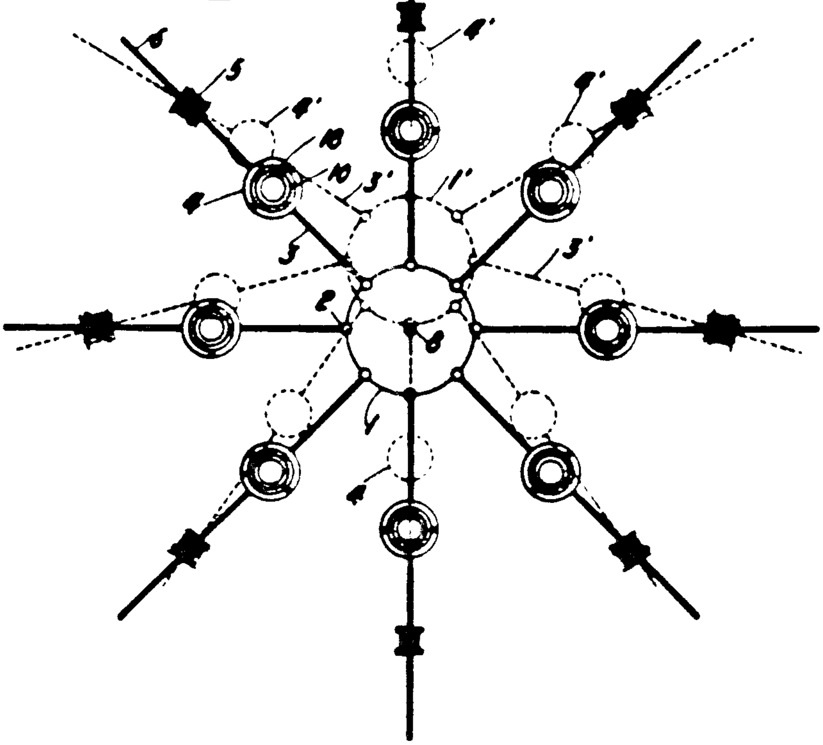


Fig. 1



June 7, 1960 H. COANDA **2,939,654**

DEVICE FOR THE SIMULTANEOUS CONTROL OF
LIFTING AND DIRECTIONAL ELEMENTS

Filed April 30, 1957 3 Sheets-Sheet 2

Fig. 5

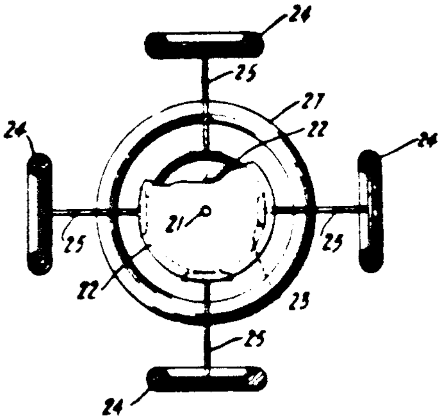


Fig. 6

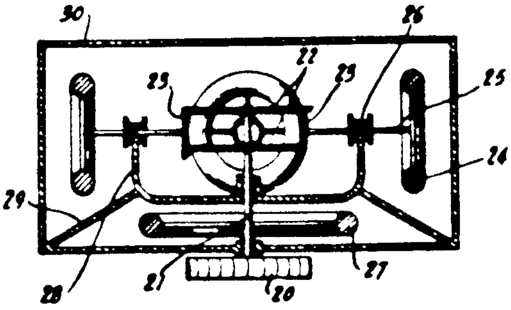
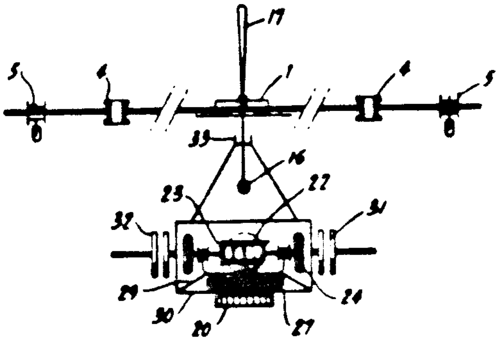


Fig. 7



June 7, 1960 H. COANDA **2,939,654**

DEVICE FOR THE SIMULTANEOUS CONTROL OF
LIFTING AND DIRECTIONAL ELEMENTS

Filed April 30, 1957 3 Sheets-Sheet 3

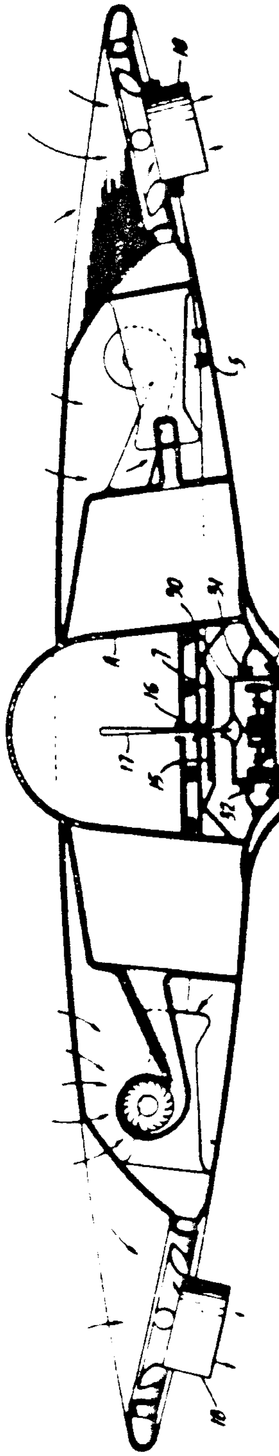


Fig. 8

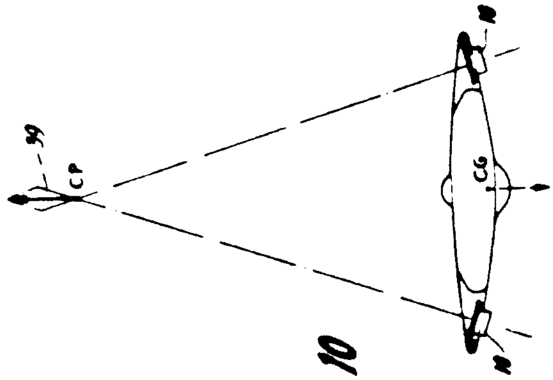


Fig. 10

JET TUBE POSITIONS
FOR TAKE-OFF

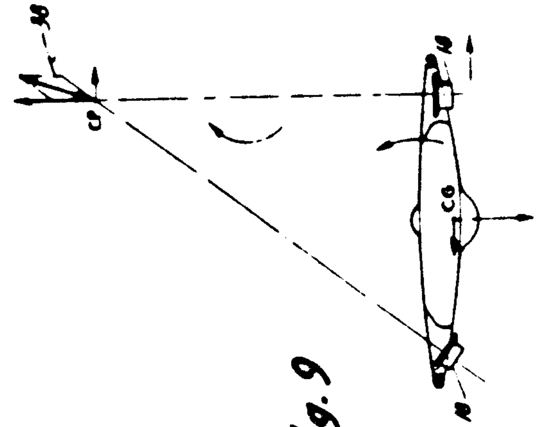


Fig. 9

JET TUBE POSITIONS
WHILE BANKING

United States Patent Office

2,939,654
Patented June 7, 1960

2,939,654

DEVICE FOR THE SIMULTANEOUS CONTROL OF LIFTING AND DIRECTIONAL ELEMENTS

Henri Coanda, Paris, France, assignor to Sebac Nouvelle S.A., a corporation of Switzerland

Filed Apr. 30, 1957, Ser. No. 655,972

Claims priority, application France Sept. 13, 1956

5 Claims. (Cl. 244—79)

In an aircraft which rises vertically (or almost vertically) by virtue of the simultaneous action of a number of elements such as discharge nozzles, rockets or similar elements, it is essential to be able to control constantly the action of these discharge nozzles or rockets and especially the direction and the force of their action, and this for a number of reasons, for example when it is desired to move the aircraft or when one of the discharge nozzles or rockets operates differently from the others, or again when the ambient medium acts in an unusual manner on the aircraft.

In addition, it happens that very rapid shocks are produced, and in this case, the corrections must be instantaneous and in consequence automatic.

It is for these reasons that the present invention has for its object a control device for lifting and directional members which can be operated either at will by the pilot or automatically from a gyroscopic unit.

A control device in accordance with the invention is applied for example to an aircraft provided with a number of discharge nozzles located along the circumference of a circle and in which the reaction directions, orientated upwards, meet each other and cross above the centre of gravity, said discharge nozzles being suspended by a swivel joint of the Cardan type.

The present invention contemplates the provision of a simultaneous control for all these nozzles, giving to each nozzle the corresponding inclination desired.

In accordance with the invention the manual or automatic control device comprises a horizontal central plate having at its periphery a number of points of articulation equal to the number of nozzles to be controlled, crank-arms fixed to the central plate joining each of the nozzles to the articulation points referred to. On these crank-arms, the fixing of the discharge nozzles is effected by joints or pivotal attachments which operate along two axes located in the same plane as the central plate, the joints each comprising at its center a sleeve member through which the nozzle to be controlled is arranged to slide, each crank-arm is extended beyond the corresponding nozzle so as to be supported by a slide situated in the plane of the central plate, each slide being articulated in its turn about a vertical axis; finally, the central plate is actuated by a lever which can pivot about a point arranged below or above the plate, the said lever being connected to the plate by a member of the same type as that which effects the control of each discharge nozzle, that is to say by a member comprising a joint of the Cardan type with a sleeve at its center.

In addition, in order to obtain automatic control, there may be provided a gyroscopic system formed on the one hand by gyroscopes having their axes in the horizontal plane and, on the other hand, by one or a number of gyroscopes placed in such manner that their axes are in a vertical plane, this gyroscopic system forming a supporting and reference trihedron in space for the aircraft.

The gyroscopic system is preferably connected to the aircraft by means of clutches. When the gyroscopic system is completely de-clutched from the whole of the aircraft, it may be coupled either directly or through a servo-mechanism to the handle of the control lever of the central plate, and from that moment the aircraft is controlled automatically by this gyroscopic system.

In accordance with a special feature of the invention, it is also possible to effect the control of the aircraft by the pilot, by coupling the gyroscopic system to the aircraft, or to turn the aircraft over to automatic control by de-clutching the gyroscopic system.

In certain forms of construction, a supplementary gyroscopic system may be provided to replace only the action of the pilot, the gyroscopic system first mentioned remaining fixed to the aircraft.

There will now be described a form of embodiment of the invention given by way of example and without any limiting sense, reference being made to the accompanying diagrammatic drawings, in which:

Fig. 1, is a diagrammatic view in plan of the whole of the control device;

Fig. 2 is an elevation view of the device in Fig. 1;

Fig. 3 is a plan view showing the articulation which controls a discharge nozzle;

Fig. 4, is a diagrammatic view showing the position of a discharge nozzle displaced by the movement given to the central plate;

Figs. 5 and 6, are diagrammatic views respectively in plan and in elevation, showing a gyroscopic system suitable for carrying the invention into effect;

Fig. 7 is a diagrammatic view showing the assembly of the control device shown in Figs. 1 to 4 with the gyroscopic system of Figs. 5 and 6;

Fig. 8, is an elevation section of an aircraft illustrating the manner in which a device or system according to the invention is mounted therein;

Figs. 9 and 10, are diagrammatic views of the manner in which the nozzles or jet tubes are positioned according to the invention for banking and take-off respectively.

Referring first of all to Figs. 1 to 4 inclusive, it is seen that a control device in accordance with the invention is constituted and operates as follows:

A central plate or central control member 1 is capable of taking up a number of positions under the action of a lever 17, of which one position has been shown in solid lines (position 1) and one in broken lines (position 1') in Fig. 1. Crank-arms or connecting rods articulated to the central plate at 2 then occupy the respective positions 3 and 3' (see Fig. 1) by sliding inside slides or guides 5 which are disposed radially of member 1 and can rotate about a vertical axis 7. Universally tiltable support means 4 occupy the positions 4' when the central control member is shifted to 1' (Fig. 1). In Fig. 4, there has been shown the position of the supports at 4'. The supports each comprise a ring 14 fixed to a ring 13 by two rotatable spindles 12 which can be seen on Fig. 3. The ring 14 is coupled to a sleeve 10 by two rotatable spindles 11. A discharge nozzle 18 fixed to a swivel joint 9 is arranged to slide inside the sleeve 10. It will be understood that a plurality of jet tubes or nozzles 18 forming power units of the jet type are arranged angularly spaced radially of member 1 and disposed spaced on the circumference of a circle.

By virtue of this arrangement, when the support 4 is moved to the position 4', which is not necessarily in the plane of symmetry, which is determined by the direction of the shaft 7 and the crank-arm 3, the discharge nozzle 18 can take up an inclined position in a new plane by sliding inside the sleeve 10, the displacement of the control member 1 having taken place due to the action of the lever 17, which is itself fixed to the aircraft by a swivel joint 16 (Fig. 8). This lever is fixed to the member 1 by an articulation 8 which is of the same kind as that which controls the discharge nozzles 18, and which is shown in Fig. 3. Finally, it may be noted that the member 1 slides on balls 15 resting on a surface 19 fixed to the aircraft.

Figs. 5 and 6 show a gyroscopic system which can be associated with advantage with the control device shown in Figs. 1 to 4. This system comprises a turbine 20 (preferably driven by the suction of the surrounding air in the apparatuses which supply the lifting or directional discharge nozzles 18 of Fig. 2) which drives a shaft 21, actuating in its turn a pair of bevel gears 22 which transmit movement to the bevel-gears 23 which drive in turn drive spindles 25 and thus transmit to gyroscopes 24 their movement of rotation. The spindles 25 rotate in bearings 26 which transmit the gyroscopic effect of the spindles 25, through the intermediary of a support 28, 29, to the whole assembly or frame 30.

In addition, on the shaft 21 is mounted a gyroscope 27, which has the effect of keeping the whole of the gyroscopic system vertical.

In Fig. 7, there has been shown an assembly of the gyroscopic system of Figs. 5 and 6 which provides for its action on the lever of the control device of Figs. 1 to 4. A set of clutches 31 and 32 couples this gyroscopic system to the aircraft. When this clutch system is released from the aircraft, it is engaged at 33 to the control lever 17 of the member 1, and in consequence, the aircraft is free to move, but the actions due to the movements which the aircraft would tend to create with respect to the central members 1 are instantaneously opposed by the control of the discharge nozzle effected (by the control device in accordance with the invention) in the opposite sense, and the corrections are automatic.

There is thus obtained an automatic control which maintains the aircraft in the position which it has previously been given.

Devices according to the present invention are particularly suitable for the control of aircraft of the type described and claimed in my co-pending application, Serial No. 626,026 filed of even date.

As shown in Fig. 8, in which the devices of the invention have been given the same reference numbers as in the preceding figures and is shown mounted in cabin A, the movement of a housing 30 for the gyroscopic device 22–25 causes the lower part of the control stick 17 to pivot about its fixed articulation 16, which causes a sliding movement of the central plate 1 mounted on balls 15. The gyroscope housing 30 can move in a pendular manner about the same pivotal point 16, the casing being supported on plates 35 rolling on balls 36 on a spherical surface 34 of the aircraft, the center of which is the center of gravity of the aircraft. Fig. 8 also shows the arrangement for transmission of movement of the central plate 1 to control the tilt of the discharge nozzles 18.

As can be seen diagrammatically in Fig. 9, the nozzles 18 are tilted in unison to bank the aircraft and a projection of the nozzle center lines corresponding to their effective lines of force intersect at CP with a resultant vector 38 corresponding to the resultant force vector making it possible to bank and thus turn. The position of the nozzles or jet tubes 18 for take-off is shown in Fig. 10. The resultant force is diagrammatically represented as a vector 39 shown as a vertical lift vector directly above the aircraft center of gravity C.G.

It will be understood that various modifications or improvements may be made to the forms of embodiment described and shown, by replacing equivalent members, without thereby departing from the spirit or from the scope of the appended claims.

What I claim is:

1. In an aircraft, a plurality of jet tubes forming power units of the jet type angularly spaced on the circumference of a circle, a source of jet power for the tubes, universally tiltable support means for tiltably supporting the jet tubes individually, a control arrangement for tilting the jet tubes simultaneously with respect to the vertical and to return them to a predetermined position with the jets discharging downwardly comprising, a central control member shiftable a predetermined extent in any direction, a plurality of angularly spaced connecting rods extending radially from the central member and each pivotally connected thereto, each rod being connected to a corresponding jet tube support for moving it to tilt an associated jet tube, for each rod a pivotally mounted guide, the guides being disposed radially of the central control member in fixed angularly spaced positions, and means for shifting the central control member in any direction within the predetermined extent.

2. In an aircraft, a plurality of jet tubes forming power units of the jet type angularly spaced on the circumference of a circle, a source of jet power for the tubes, universally tiltable support means for tiltably supporting the jet tubes individually, a control arrangement for tilting the jet tubes simultaneously with respect to the vertical and to return them to a predetermined position with the jets discharging downwardly comprising, a central control member shiftable a predetermined extent in any direction, a plurality of angularly spaced connecting rods extending radially from the central member and each pivotally connected thereto, each rod being connected to a corresponding jet tube support for moving it to tilt an associated jet tube, for each rod a pivotally mounted guide, the guides being disposed radially of the central control member in fixed angularly spaced positions, and manual means for shifting the central control member at will in any direction within the predetermined extent.

3. In an aircraft, a plurality of jet tubes forming power units of the jet type angularly spaced on the circumference of a circle, a source of jet power for the tubes, universally tiltable support means for tiltably supporting the jet tubes individually, a control arrangement for tilting the jet tubes simultaneously with respect to the vertical and to return them to a predetermined position with the jets discharging downwardly comprising, a central control member shiftable a predetermined extent in any direction, a plurality of angularly spaced connecting rods extending radially from the central member and each pivotally connected thereto, each rod being connected to a corresponding jet tube support for moving it to tilt an associated jet tube, for each rod a pivotally mounted guide, the guides being disposed radially of central control member in fixed angularly spaced positions, and means comprising a lever manually operable by a pilot for shifting the central control member in any direction with the predetermined extent.

4. In an aircraft, a plurality of jet tubes forming power units of the jet type angularly spaced on the circumference of a circle, a source of jet power for the tubes, universally tiltable support means for tiltably supporting the jet tubes individually, a control arrangement for tilting the jet tubes simultaneously with respect to the vertical and to return them to a predetermined position with the jets discharging downwardly comprising, a central control member shiftable a predetermined extent in any direction, a plurality of angularly spaced connecting rods extending radially from the central member and each pivotally connected thereto, each rod being connected to a corresponding jet tube support for moving it to tilt an associated jet tube, for each rod a pivotally mounted guide for slidably guiding the rods the guides being disposed radially of the central control member and radially outwardly of the jet tubes in fixed angularly spaced positions, and means for shifting the central control member in any direction within the predetermined extent.

5. In an aircraft, a plurality of jet tubes forming power units of the jet type angularly spaced on the circumfer-

ence of a circle, a source of jet power for the tubes, universally tiltable support means for tiltably supporting the jet tubes individually, a control arrangement for tilting the jet tubes simultaneously with respect to the vertical and to return them to a predetermined position with the jets discharging downwardly comprising, a central control member shiftable a predetermined extent in any direction, a plurality of angularly spaced connecting rods extending radially from the central member and each pivotally connected thereto, each rod being connected to a corresponding jet tube support for moving it to tilt an associated jet tube, for each rod a pivotally mounted guide, the guides being disposed radially of the central control member in fixed angularly spaced positions, and gyroscopically controlled means for automatically shifting the central control member in any direction within the predetermined extent for maintaining a preset attitude of the aircraft.

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LIFTING APPARATUS

Filed May 20, 1964 5 Sheets-Sheet 1

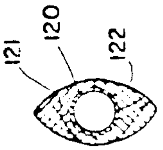


FIG. 3

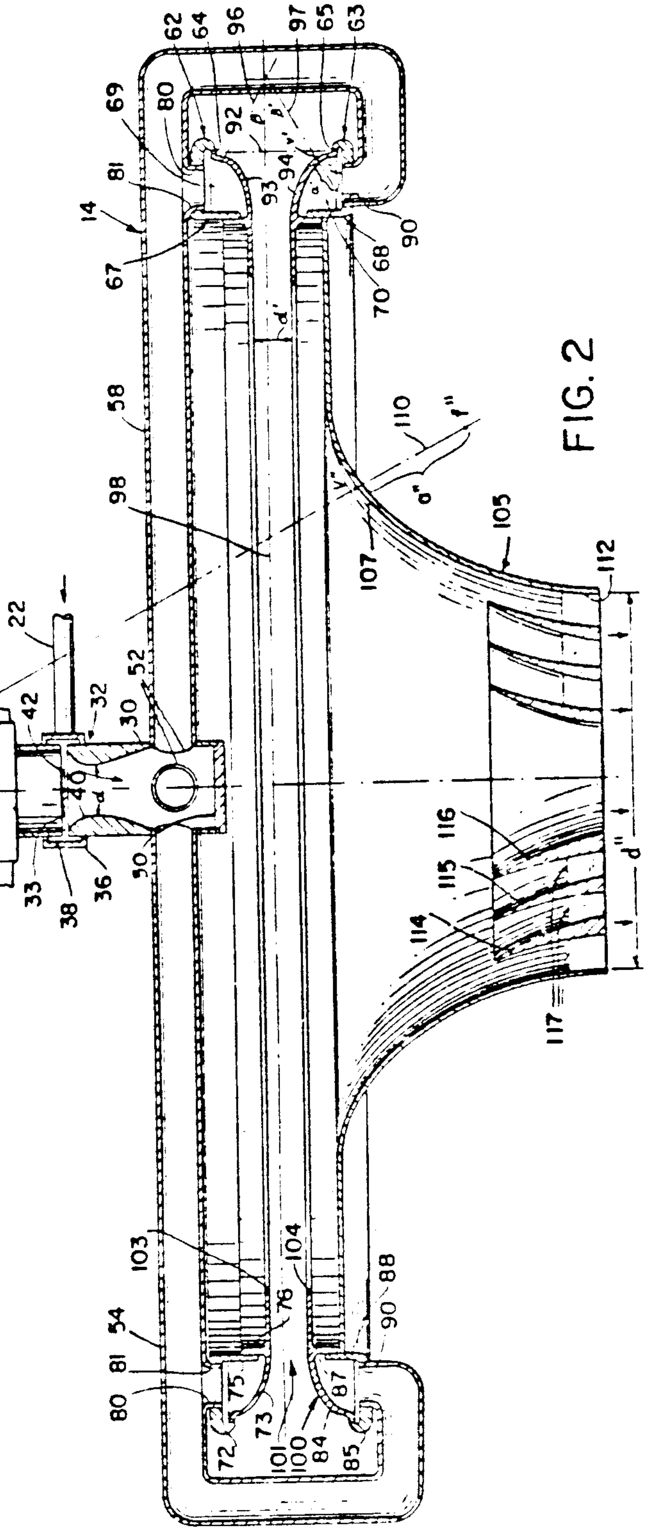
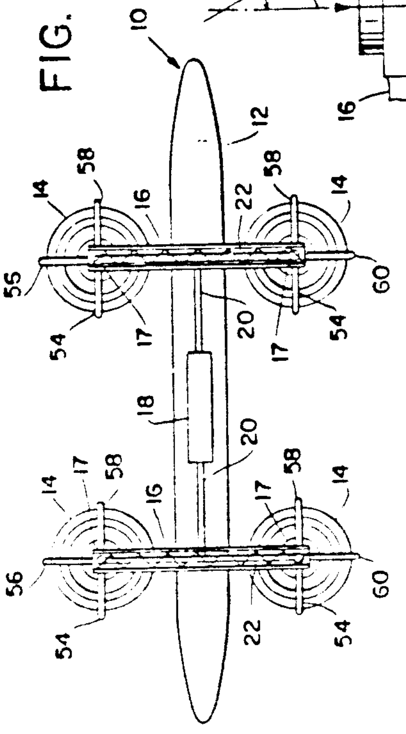


FIG. 2

FIG. 1



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LIFTING APPARATUS

Filed May 20, 1964 5 Sheets-Sheet 2

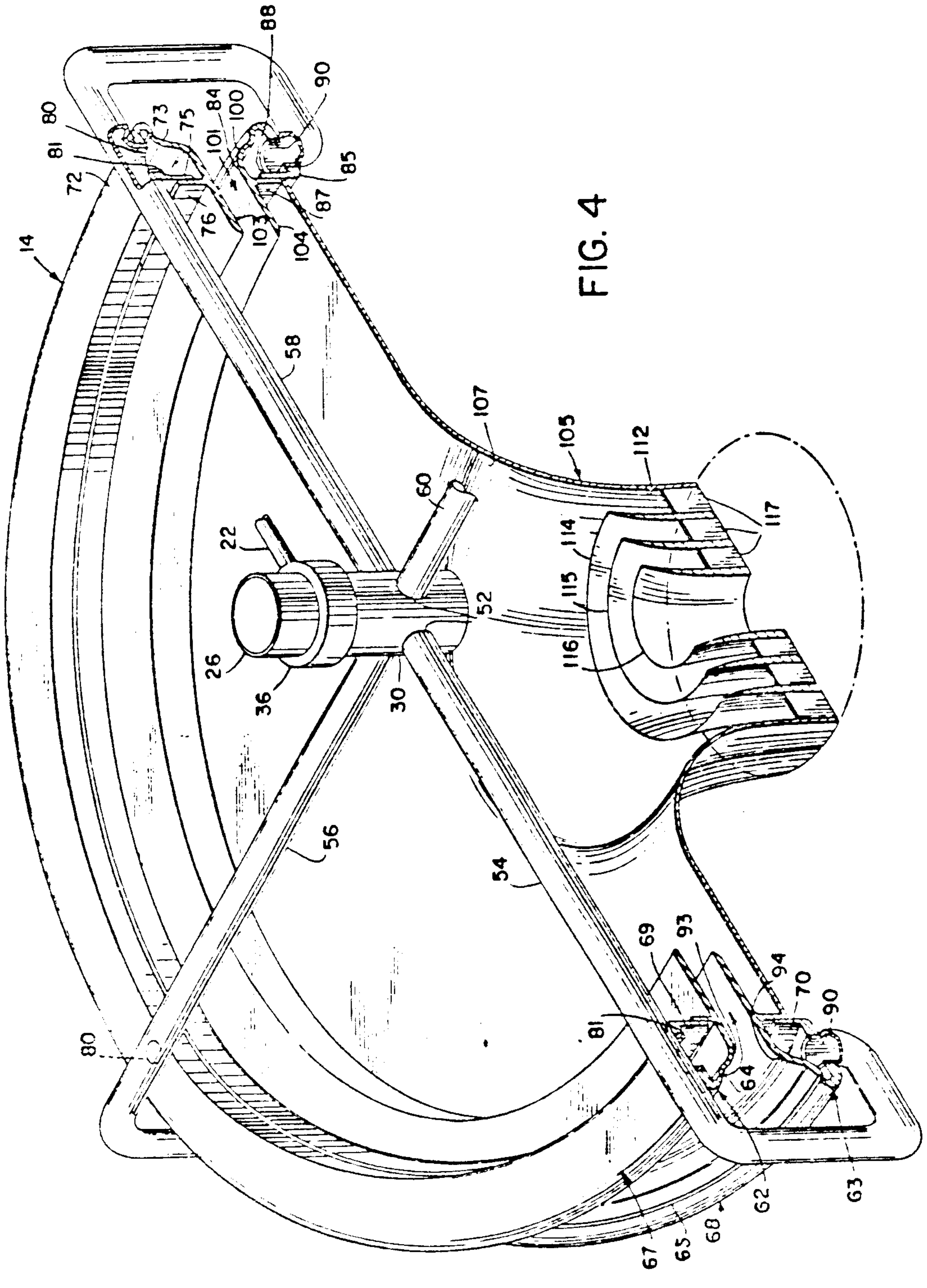

FIG. 4

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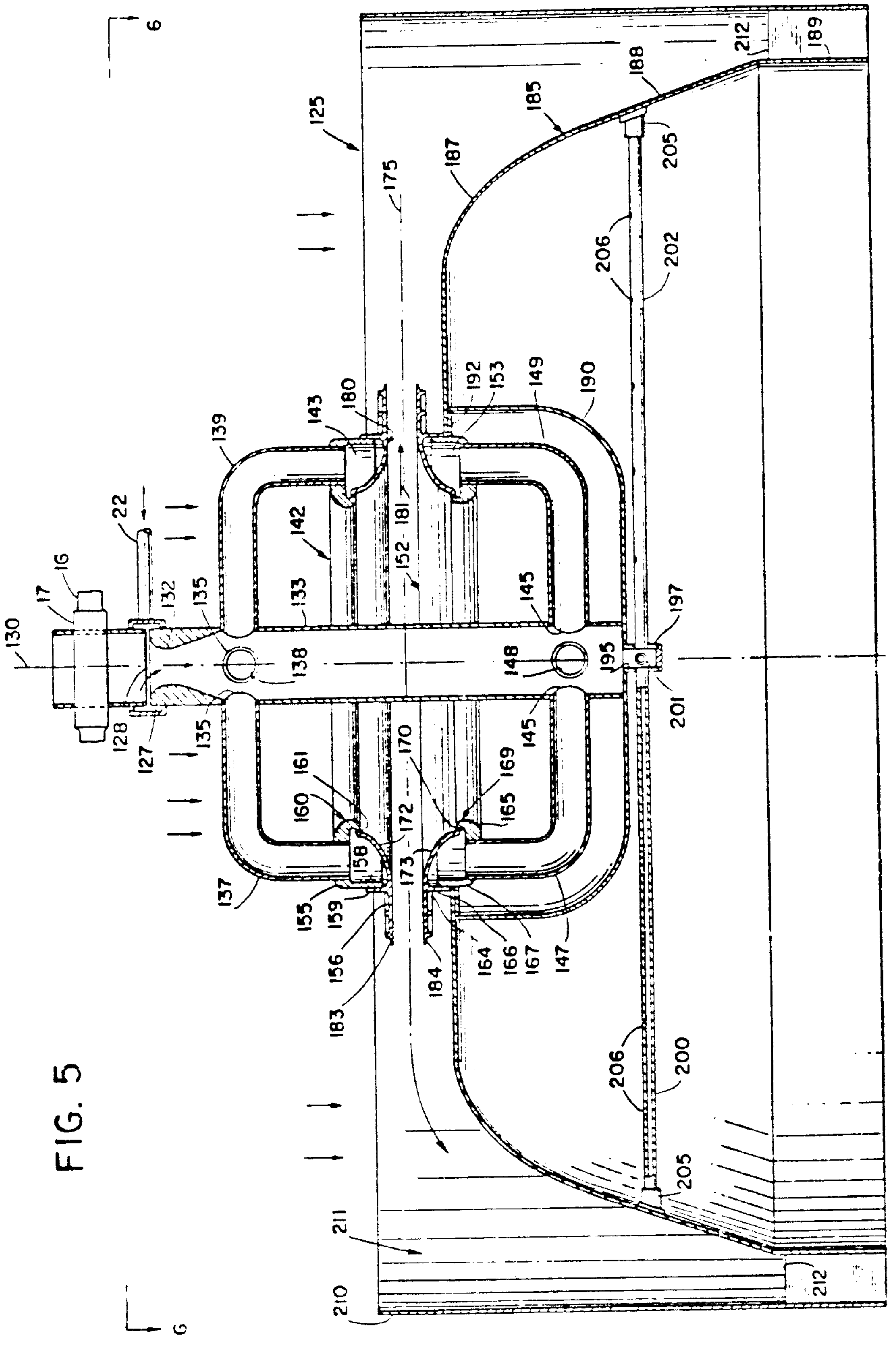

FIG. 5

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LIFTING APPARATUS

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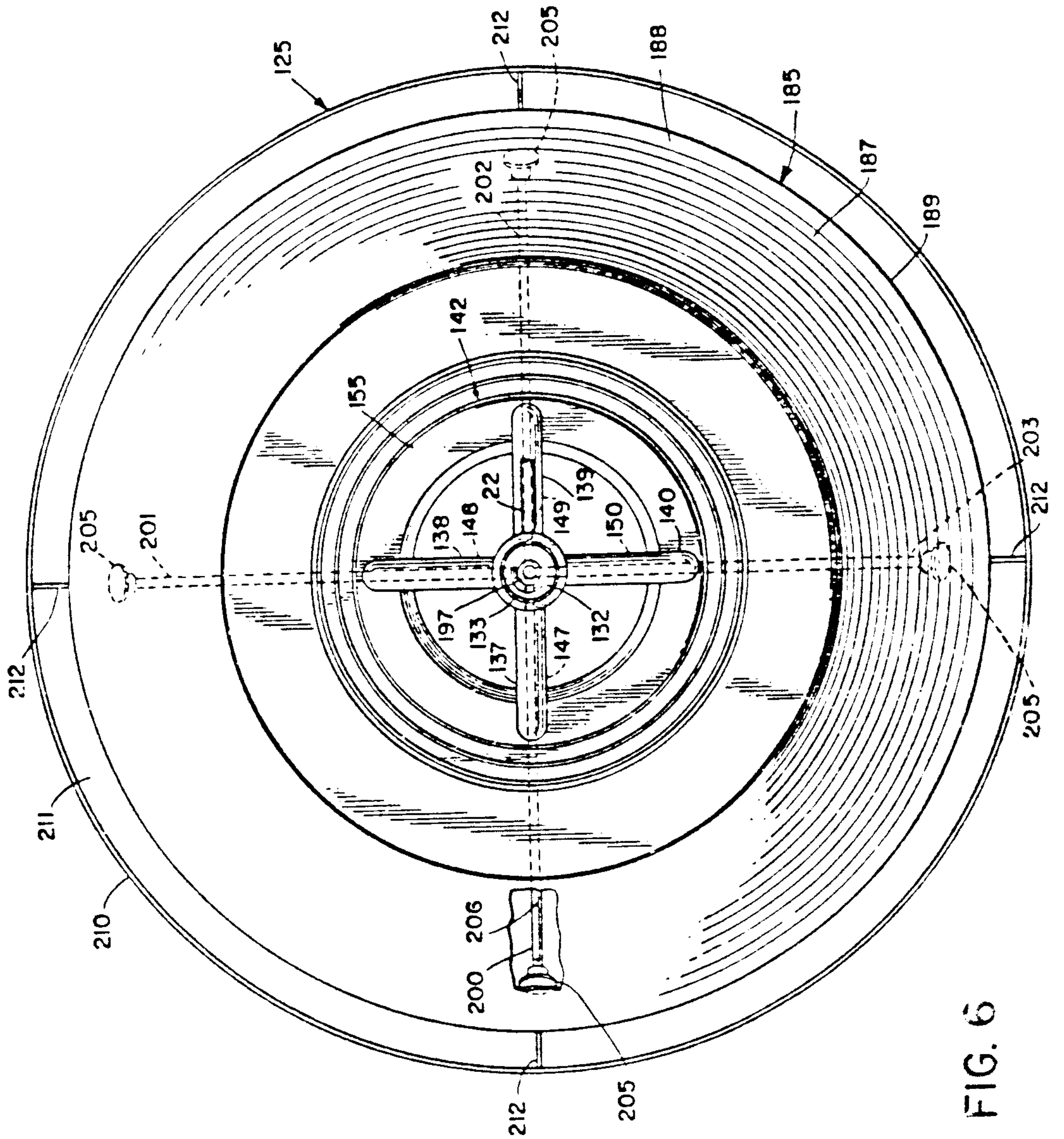

FIG. 6

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LIFTING APPARATUS

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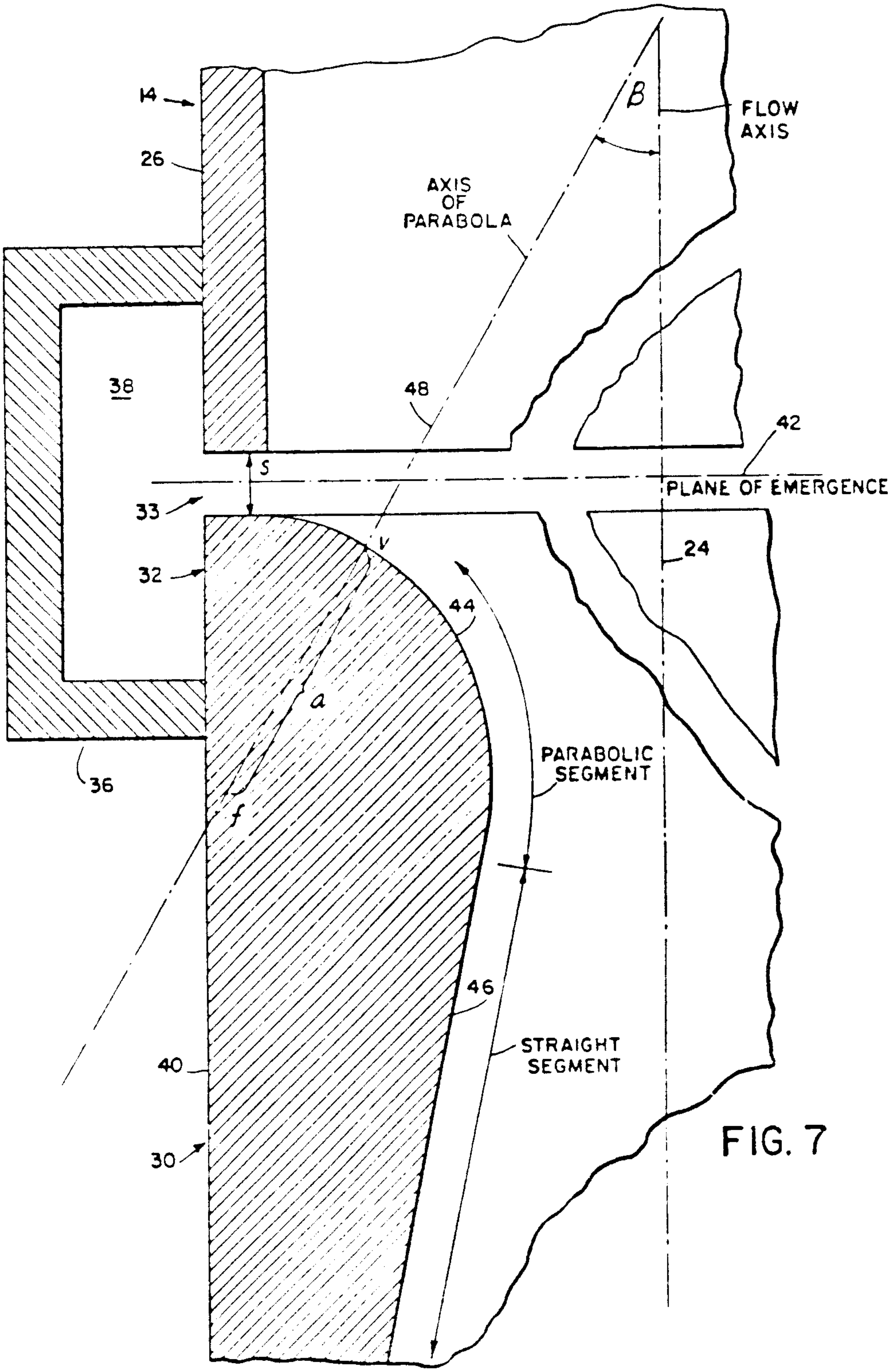

FIG. 7

United States Patent Office

3,261,162
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3,261,162
LIFTING APPARATUS
Henri Coanda, Paris, France
(777 Summer St., Stamford, Conn.)
Filed May 20, 1964, Ser. No. 368,852
20 Claims. (Cl. 60—35.5)

This invention relates to lifting apparatus and more particularly to such apparatus which utilizes the physical phenomenon known as the Coanda effect.

The present invention, while of general application, is particularly well suited for use as a propulsion device for aircraft and similar vehicles of the so-called Vertical Take Off-Landing (VTOL) type. Such aircraft are capable of moving both vertically and horizontally and of hovering in a substantially stationary position with a high degree of inherent stability.

In one form of prior apparatus of this type, the lift and horizontal thrust for the aircraft are produced through the use of a series of blower devices which are mounted on a support of generally discoid configuration. As more fully described in U.S. Patent 2,988,303, granted June 13, 1961, to Henri Coanda, for example, the blower devices are arranged radially with respect to the apparatus and direct entrained air along spiral paths to groups of jet discharge nozzles. Each device is provided with one or more slots which utilize the Coanda effect to entrain a mass of ambient air. The air is entrained from one side of the apparatus, while the discharge nozzles are arranged on the opposite side, thus producing a differential pressure to cause motion in the direction of the lower pressure.

As disclosed, for example, in U.S. Patent 2,052,869, granted September 1, 1936, to Henri Coanda, the Coanda effect is produced when air or other fluid under pressure leaves a chamber through an orifice disposed adjacent an extended lip which continuously recedes from the direction of emergence of the fluid. The stream of fluid which passes through the orifice has a tendency to follow the surface having the extended lip, carrying with it a supplementary mass of the ambient fluid. In this way a large volume of ambient fluid is entrained and directed by a small mass of fluid under pressure.

Heretofore, propulsion apparatus of the foregoing type have exhibited certain disadvantages. As an illustration, for apparatus of the type previously employed which was capable of producing a given lift, for example, efforts to increase the lift by increasing the size of the apparatus seriously affected the operating efficiency. As a result, particularly for aircraft traveling at relatively high altitudes, such prior apparatus frequently proved deficient in the realization of the necessary lift. In addition, the comparatively large number of blower devices employed in apparatus of the type used heretofore, together with the various spiral passageways, etc., often necessitated the use of expensive and complicated piping equipment. These latter difficulties have been further compounded by the need for directing the output of the blowers to the jet discharge nozzles.

One general object of this invention, therefore, is to provide new and improved apparatus for producing a fluid propulsion force.

Another object of the invention is to provide propulsion apparatus for aircraft and related vehicles.

Still another object of the invention is to provide apparatus of the character indicated which makes extremely effective use of the Coanda effect.

A further object of this invention is to provide such apparatus which enables the realization of substantially greater lift than has been obtainable heretofore.

Still another object of the invention is to provide lift

producing apparatus which is economical to manufacture and thoroughly reliable in operation.

In one illustrative embodiment of this invention, there is provided propulsion apparatus of generally lenticular configuration which includes a pair of fluid conductor means which are symmetrically disposed about an axis of flow. The fluid conductor means respectively include first and second Coanda slots, and their facing surfaces from a third Coanda slot. Fluid under pressure is directed to the fluid conductor means and simultaneously through the first and second Coanda slots to induce a primary flow of ambient air into the third slot. As the air passes through this latter slot, a secondary ambient flow is induced into the stream. The secondary flow is derived from the upper surface of the apparatus, thus providing a zone of reduced pressure thereadjacent and the combined flow follows the direction of the flow axis to provide a zone of increased pressure beneath the apparatus.

In accordance with one feature of the invention, the secondary flow of ambient fluid, after passing through the third Coanda slot, is directed over the curved surface of a lip member which is supported on the downstream side of the third slot. In several advantageous embodiments, the third slot is arranged in cascading relationship with the first and second slots, and the throat dimensions for the various slots, rather than being directly proportional to the overall size of the apparatus, are established in accordance with unique and predetermined relationships to provide optimum operating efficiency for apparatus of widely varying size. In addition, the curved surface of the lip member is of particular configuration and is defined by a parabolic segment which is revolved about the axis of the apparatus to enable maximum effective utilization of the Coanda effect.

In accordance with another feature of the invention, in several good arrangements, each of the fluid conductor means comprises an annular conduit having a curved surface which is parabolic in shape and which cooperates with the corresponding first or second Coanda slot. The various parabolic surfaces adjacent the Coanda slots are arranged to provide a substantial pressure differential between the upper and lower faces of the apparatus to produce an unusually high lift. Moreover, particularly because of the momentum produced by the action of the annular conduits, the apparatus enables the realization of an extremely large flow augmentation.

In accordance with a further feature of certain embodiments of the invention, there is provided a unique arrangement for supplying fluid to the annular conduits. The arrangement includes an additional Coanda slot which induces a substantial flow of ambient air into the conduits from adjacent the upper face of the apparatus. With this arrangement, the lift and flow augmentation are further enhanced.

The present invention, as well as further objects and features thereof, will be understood more clearly and fully from the following description of certain preferred embodiments, when read with reference to the accompanying drawings, in which:

FIGURE 1 is a schematic plan view of an aircraft utilizing a plurality of propulsion apparatus in accordance with one preferred embodiment of this invention;

FIGURE 2 is an enlarged vertical cross-sectional view, with certain parts broken away, of one of the propulsion apparatus shown in FIGURE 1;

FIGURE 3 is a vertical cross-sectional view of an airfoil and associated parts useful with the apparatus;

FIGURE 4 is a perspective view, partially broken away and in section, of the apparatus illustrated in FIGURE 2;

FIGURE 5 is a vertical cross-sectional view, with certain parts broken away, of propulsion apparatus in accordance with another preferred embodiment of the invention;

FIGURE 6 is a horizontal sectional view taken along the line 6—6 of FIGURE 5, with certain portions broken away to show internal structure; and

FIGURE 7 is an enlarged cross-sectional view, partially schematic and with certain parts broken away, of a portion of the apparatus of FIGURE 2.

Referring to FIGURE 1, an aircraft 10 comprises a fuselage schematically shown at 12 and four propulsion devices 14 of generally discoid configuration. In the illustrated embodiment, the devices 14 are mounted at the outer ends of two elongated structural members 16 which are transversely supported in spaced-apart relationship with each other on the upper surface of the fuselage 12. A gimbal connection 17 serves to maintain each of the devices 14 pivotally suspended from the corresponding structural member to enable movement of the devices (by means not shown) to control the direction of travel of the aricraft. In other good arrangements, the devices 14 are mounted in stationary positions, such as within a wing or other airfoil, for example, and the direction of flight is determined by movable vanes carried therebeneath.

Each of the propulsion devices 14 is supplied with fluid under pressure from a pressure source 18. In the embodiment of FIGURE 1, the source 18 advantageously comprises a gas turbine for generating a pressurized gaseous fluid, although in other embodiments liquids such as water, etc., may be employed with good effect. The source 18 is arranged to supply the fluid to each of the devices 14 through a conduit 20 and a branch conduit 22.

As best shown in FIGURE 2, each of the propulsion devices 14 is uniformly disposed about a centrally located flow axis 24. The gimbal connection 17, shown schematically in this figure, supports an upstanding tube 26 which is positioned along the axis 24 and is open at its upper end to the atmosphere. The lower end of the tube 26 is in spaced juxtaposition with a second tube 30 which serves as an internal nozzle and is coaxially aligned with the tube 26. The facing portions of the tubes 26 and 30 form a Coanda nozzle 32 having an annular Coanda slot 33 therebetween. The nozzle 32 is enclosed by a cylindrical jacket 36 having an annular chamber 38 therein which is in open communication with the corresponding branch conduit 22. The pressurized fluid from the conduit 22 is introduced into the chamber 38 and emerges from the slot 33 in a horizontal plane indicated schematically in FIGURE 2 by the line 42. This plane meets the flow axis 24 at a right angle.

The characteristics of the Coanda nozzle 32, which in some respects are similar to those of the other Coanda nozzles in the various illustrated embodiments of the invention, can best be understood with reference to FIGURES 2 and 7. The nozzle 32 includes a protruding lip 40 which is formed on the inner cylindrical surface of the tube 30 a short distance beneath the plane of emergence 42 of the pressurized fluid through the Coanda slot 33. The inner surface of the tube 30 converges sharply to a throat having a minimum planar area of a diameter "*d*" and then continuously increases in size, thus forming a convergent-divergent nozzle.

FIGURE 7 is an enlarged, partially schematic view of a portion of FIGURE 2. The lip 40 includes a parabolic surface 44 which extends from immediately adjacent the Coanda slot 33 to a point adjacent the throat of the tube 30. This portion of the lip 40 continuously recedes from the plane 42 of emergence of the pressurized fluid. The surface 44 is formed by a segment of a parabola which is removed about the flow axis 24 and includes a parabolic axis 48. This latter axis intersects the surface 44 at a vertex v and advantageously forms an acute angle β (beta) with respect to the direction of fluid flow along the axis 24 to achieve the optimum Coanda effect. A comparatively straight segment 46 merges with the parabolic segment and defines the diverging portion of the lip 40.

In some embodiments of the invention, the angle β advantageously lies within predetermined, well defined limits and is determined in part by the maximum desired lift in a particular propulsion device. In cases in which comparatively high maximum lift is needed, the angle β is small, while somewhat larger angles are employed in cases in which the lift requirements are lower. In certain particularly good arrangements, the angle β is within the range of from about 10° to about 45° to achieve the optimum lift characteristics.

The vertex v of the parabola forming the parabolic surface 44 is located on this surface in spaced relationship with the plane 42 of emergence. The shape of the surface 44 may be expressed in terms of the distance a between the vertex v and the parabolic focus f lying along the axis 48. In several good embodiments, this distance a is closely related to the throat diameter d (FIGURE 2) such that the ratio d/a lies within a predetermined range. Preferably, the ratio extends between 5 and 25 to provide a Coanda lip 40 of optimum shape and thereby make maximum effective use of the Coanda effect.

As the pressurized fluid from the source 18 (FIGURE 1) passes through the Coanda solt 33, a low pressure zone is formed along the parabolic surface 44 of the lip 40. Because of the annular configuration of this surface, the low pressure zone extends across the tube 30 immediately adjacent the throat and entrains a large mass of ambient air from the open upper end of the tube. The combined air and fluid move vertically along the flow axis 24 at reduced pressure and substantially increased velocity. The Coanda nozzle 32 and associated parts thus serve as the first stage of the device 14 and produce a primary flow of accelerated ambient air which is taken from the upper portion of the device and passes through the tube 30 along a downwardly directed path.

The lower end of the tube 30 is closed by a generally circular plate 50. As best shown in FIGURES 2 and 4, four circular openings 52 are provided in the tube immediately above the plate 50 and are respectively connected to radially extending pipes 54, 56, 58 and 60. These pipes are arranged in a horizontal plane at right angles to each other, the pipes 54 and 58 being coaxially supported and extending in a direction perpendicular to the coaxial pipes 56 and 60.

There are two second stage Coanda nozzles 62 and 63 and corresponding Coanda slots 64 and 65. The slots 64 and 65 are respectively defined by two fluid conductor means, such as the annular conduits 67 and 68, for example, which enclose corresponding fluid chambers 69 and 70. The conduits 67 and 68 are symmetrically disposed about the axis of flow 24 and are maintained in spaced-apart relationship with each other in a manner that will become more fully apparent hereinafter.

The annular conduit 67 includes an upper ring 72 and a lower ring 73. The ring 72 is provided with an integrally formed depending portion 75 which is rigidly affixed to an upwardly protruding portion 76 of the ring 73 to form the inner side wall of the conduit 67. The outermost rim of the ring 72 is bent downwardly and is positioned in spaced relationship with the adjacent rim of the ring 73 such that the facing portions of the rings form the upper Coanda nozzle 62, the space therebetween being of annular configuration and defining the Coanda slot 64. This slot connects the chamber 69 to the atmosphere and faces in a generally downward direction. Four circular openings 80 are formed in the upper surface of the ring 72 at ninety-degree intervals to respectively accommodate depending tubes 81 integrally formed with the radially extending pipes 54, 56, 58 and 60. The tubes 81 provide support for the annular conduit 67 and serve to maintain each pipe in open communication with the chamber 69.

In a similar manner, the annular conduit 68 is fabricated from an upper ring 84 and a lower ring 85. These rings are connected together by respective integrally formed portions 87 and 88 which serve as the inner wall of the chamber 70. The outer rim of the ring 85 is bent upwardly and is arranged in spaced relationship with the adjacent rim of the ring 84 to form the Coanda slot 65 therebetween, and the facing portions of the rings 84 and 85 define the Coanda nozzle 63. The slot 65 faces upwardly and connects the chamber 70 to the atmosphere. The ring 85 includes four openings 90 at ninety-degree intervals which respectively accommodate the free ends of the pipes 54, 56, 58 and 60, thus providing support for the conduit 68 and maintaining the same in spaced relationship with the conduit 67. The outermost portions of these pipes extend downwardly and are of generally U-shaped configuration to meet the openings 90.

The accelerated initial flow of ambient air, together with the pressurized fluid from the source 13 (FIGURE [illegible] moves through the pipes 54, 56, 58 and 60 and into the chamber 69 formed by the upper conduit 67 and the chamber 70 formed by the lower conduit 68. The accelerated fluid then passes simultaneously through the Coanda slots 64 and 65. These slots are in vertical alignment with each other such that the fluid passing through the slots emerges in a direction parallel to the flow axis 24, as shown schematically in FIGURE 2 by the line 92, with the fluid from the slot 64 being directed in a downward direction while that from the slot 65 being directed upwardly.

On the downstream side of the fluid emerging from the Coanda slot 64, there is provided an external parabolic surface 93 on the lower ring 73 of the conduit 67. Similarly, the upper ring 84 of the conduit 68 includes an external parabolic surface 94 on the downstream side of the fluid emerging from the Coanda slot 65. The surfaces 93 and 94 form protruding lips for the Coanda nozzles 62 and 63, respectively, and are each defined by a segment of a parabola which is revolved about the main flow axis 24. As the fluid in the conduits 67 and 68 passes through the slots 64 and 65, a low pressure zone is formed along the surfaces 93 and 94 which extends throughout the space between the conduits. A large mass of secondary ambient air is entrained from the peripheral edge of the device which merges with the fluid from the slot 64 and 65. The combined air and fluid move horizontally in a generally radial direction toward the flow axis 24 along a plane indicated schematically at 98.

The parabolic axis 96 for the surface 93 and the parabolic axis 97 for the surface 94 each form an acute angle β' with respect to the direction 98 of the fluid flowing between the conduits 67 and 68. Although various angles β' may be employed in accordance with the invention, depending among other things on the maximum desired lift for the device, in some advantageous embodiments the angle β' preferably is within the 10° to 45° range specified above for best results.

As indicated heretofore, the shape of a given parabolic surface may be expressed in terms of the distance between the vertex and the parabolic focus. With respect to the parabolic surfaces 93 and 94, the vertex v' and the focus f' are separated by a distance a'. This distance is closely related to the minimum throat distance d' between the annular conduits 67 and 68. In certain advantageous arrangements, the ratio d'/a' extends between 5 and 25 for each of the surfaces 93 and 94. By maintaining the ratio within this range, a substantial increase in momentum is imparted to the air moving inwardly from the conduits 67 and 68 to enable the realization of an extremely large propulsion force.

The facing portions of the annular conduits 67 and 68 form an additional Coanda nozzle 100, the space between the conduits defining a Coanda slot 101. The nozzle 100 and the slot 101 comprise the third stage of the device and are formed by the facing parabolic surfaces 93 and 94 and by two ring-shaped flanges 103 and 104 which are integrally connected to the rings 73 and 85, respectively, and extend inwardly from the corresponding surfaces 93 and 94. The flanges 103 and 104 serve as substantially straight extensions of the surfaces.

Rigidly affixed to the lower conduit 68 is an annular lip member 105 which is symmetrically disposed about the flow axis 24. The member 105 is of circular cross section and includes a parabolic surface 107 which continuously recedes from the plane of emergence, represented by the line 98, of the fluid moving through the Coanda slot 101. The surface 107 is formed by a segment of a parabola which is revolved about the flow axis 24 and includes a parabolic axis 110. In a manner similar to the parabolic axis 48 described above, the axis 110 forms an acute angle β'' with respect to the direction of fluid flow along the axis 24. In several good embodiments of the invention, the angle β'' is maintained within the range of from about 10° to about 45° to achieve the optimum Coanda effect. A comparatively straight segment 112 merges with the parabolic segment adjacent the lower portion of the propulsion device.

The vertex v'' of the parabola forming the surface 107 is positioned on this surface in spaced relationship with the plane of emergence 98. The distance a'' between the vertex v'' and the parabolic focus f'' on the axis 110 is closely related to the throat diameter d''. In several advantageous arrangements, the ratio d''/a'' extends between the limits indicated heretofore to provide a surface 107 of optimum shape and thus enable the maximum effective use of the Coanda effect.

As an aid to streamlining the flow of fluid through the throat d'' of the member 105, annular vanes 114, 115 and 116 are located within the member 105 in coaxial relationship therewith. These vanes are supported in position by thin, aerodynamically shaped brackets 117.

In operation, fluid under a pressure which is illustratively of the order of one hundred pounds per square inch gauge, for example, is introduced through the conduit 22 into the annular chamber 38. The fluid passes through the Coanda slot 33 and in accordance with the Coanda effect follows the curvature of the protruding lip 40. The influx of fluid serves to reduce the pressure at the throat d below atmospheric and causes an induced flow of ambient air to be sucked into conduit 26.

The induced fluid and the fluid moving through the Coanda slot 33 flow downwardly through the tube 30, the velocity being reduced in the diverging portion of the tube to raise the pressure to just above atmospheric pressure (about 1.2 atmospheres) adjacent the pipes 54, 56, 58 and 60. The combined fluids pass through these pipes and into the annular conduits 67 and 68, where the fluids are uniformly distributed throughout the peripheral portion of the device. The fluids serve as a primary flow which passes simultaneously through the second stage annular Coanda slots 64 and 65 and in accordance with the Coanda effect follows the contour of the parabolic surfaces 93 and 94. A zone of reduced pressure is created within the throat d' to entrain a secondary flow of ambient air from adjacent the periphery of the device. Substantial momentum is imparted to this secondary flow, and a low pressure area is formed around the peripheral edge of the device which provides greatly increased stability during flight.

The combined fluids pass through the third stage Coanda slot 101 and follow the contour defined by the inner parabolic surface 107 of the member 105. The slot 101 is in cascading relationship with the second stage slots 64 and 65, that is, the combined fluids from the slots 64 and 65, together with the induced ambient flow, serve as the input for the slot 101. This input produces a reduced pressure in the vicinity of the throat d'' and causes a tertiary flow of ambient air which enters the device from adjacent its upper surface. The ambient flow, along with the air withdrawn from above the tube 26 by the first stage Coanda nozzle 32, form a low pressure area immediately above the upper surface of the device. The fluids directed downwardly along the flow axis 24 from the member 105, on the other hand, produce a zone of increased pressure immediately beneath the device. The resulting pressure differential between the upper and lower surfaces urges the device in the direction of the reduced pressure to produce substantial lift for the aircraft 10 (FIGURE 1).

FIGURE 3 is illustrative of an arrangement for reducing heat loss and air resistance across a laterally extending pipe 120. The pipe 120, which is representative of the pipes 54, 56, 58 and 60 described heretofore, includes a first insulating member 121 affixed to its upper surface and a second insulating member 122 on its lower surface. The members 121 and 122 are aerodynamically contoured and serve as fins to reduce drag. Although the members 121 and 122 may be fabricated from various insulating materials, wood is particularly effective in minimizing any heat losses that might otherwise occur.

Referring now to FIGURES 5 and 6, there is shown a fluid propulsion device 125 of generally discoid configuration which is constructed in accordance with another preferred embodiment of the invention. The device 125 includes a first stage Coanda nozzle 127 and a corresponding Coanda slot 128 which are substantially the same as the first stage nozzle 32 (FIGURE 2) and the slot 33 described above with respect to the device 14. Fluid under pressure is fed to the slot 128 from the branch conduit 22 connected to the pressure source 18 (FIGURE 1). Because of the Coanda effect, the fluid is directed downwardly along a vertical flow axis 130 past the inner parabolic surface 132 of a tube 133. This tube serves as an internal nozzle in a manner similar to the tube 30 of the device 14 and entrains a comparatively large mass of ambient air from adjacent its open upper end.

The tube 133 is substantially longer than the tube 30 of the propulsion device 14. Four openings 135 are provided in the tube 133 a short distance beneath the parabolic surface 132, and these openings are respectively connected to radially extending pipes 137, 138, 139 and 140 (FIGURE 6). The pipes 137, 138, 139 and 140 extend horizontally from the tube 133 at right angles to each other, and their free ends are bent downwardly and are fixedly secured to an annular conduit 142 having a fluid conducting chamber 143 therein. In a similar manner, the lower end of the tube 133 includes four openings 145 which respectively accommodate radially extending pipes 147, 148, 149 and 150, these pipes being hidden in FIGURE 6 by the pipes 137, 138, 139 and 140. The outer ends of the pipes 147, 148, 149 and 150 are bent in an upward direction and are connected to an annular conduit 152. This latter conduit defines a fluid conducting chamber 153.

The annular conduits 142 and 152 are symmetrically disposed about the flow axis 130 in spaced relationship with each other and are of a diameter somewhat less than that of the conduits 67 and 68 (FIGURE 2). The conduit 142 is fabricated from an upper ring 155 and a lower ring 156 which are fixedly secured together by flanges 158 and 159. These flanges form the outer peripheral wall of the chamber 143. The inner portions of the rings 155 and 156 are spaced apart and serve as a second stage Coanda nozzle 160, the space between the inner portions defining a Coanda slot 161. The slot 161 is of annular configuration and is arranged to discharge the fluid received from the pipes 137, 138, 139 and 140 in a vertically downward direction parallel to the flow axis 130.

The annular conduit 152 forms a mirror image of the conduit 142. The conduit 152 includes an upper ring 164 and a lower ring 165 which are maintained in fixed relationship with each other by flanges 166 and 167. The inner edges of the rings 164 and 165 are spaced

apart to form an additional second stage Coanda nozzle 169 and corresponding Coanda slot 170 therebetween. The fluid from the pipes 147, 148, 149 and 150 emerges from the Coanda slot 170 and initially follows an upwardly directed path parallel to the axis 130.

The ring 156 of the upper conduit 142 and the ring 164 of the lower conduit 152 respectively include parabolic surfaces 172 and 173 which serve as lips for the second stage Coanda nozzles 160 and 169. The surfaces 172 and 173 are located internally with respect to their corresponding conduits on the downstream side of the Coanda slots 161 and 170. Fluid emerging from these slots follows the path of the surfaces 172 and 173 and is directed outwardly in a radial direction transverse to the flow axis 130 along a horizontal plane 175. The fluid entrains a large mass of ambient air from adjacent the center of the device which is sucked into the space between the conduits 142 and 152.

In a manner similar to that discussed heretofore with respect to the parabolic surfaces 93 and 94 (FIGURE 2) of the propulsion device 14, the surfaces 172 and 173 on the respective conduits 142 and 152 include parabolic axes which form acute angles with respect to the direction of flow of the fluid moving between the conduits. In addition, the distances between the vertices of the parabolas and the parabolic foci are closely related to the minimum throat distance between the conduits in accordance with a predetermined ratio. Although these angles and ratios vary widely depending upon the particular characteristics of a given propulsion device, in several particularly effective embodiments the angles and ratios are maintained within the preferred ranges described above.

The facing portions of the conduits 142 and 152 form a third stage Coanda nozzle 180 having an annular Coanda slot 181. The slot 181 extends outwardly in a radial direction from the parabolic surfaces 172 and 173 between two ring-shaped flanges 183 and 184. These flanges are integrally formed with the respective rings 156 and 164 and form extensions of the corresponding parabolic surfaces. The fluid emerging from the second stage Coanda slots 161 and 170 changes direction as it passes over the parabolic surfaces in accordance with the Coanda effect and, together with the entrained ambient air, is directed horizontally through the third stage slot 181.

Surrounding the lower conduit 152 immediately beneath the Coanda slot 181 is an annular lip member 185 which serves as a skirt to control the flow of fluid through the slot. The member 185 is coaxially supported about the flow axis 130, and its inner rim is rigidly secured to the lower conduit. The member is of circular cross-section and includes an outer parabolic surface 187 which continuously recedes from the plane 175 of emergence of the fluid moving through the slot 181. The surface 187 is defined by a segment of a parabola which is revolved about the axis 130 and includes a parabolic axis which meets the direction of fluid flow along the axis 130 at an acute angle. This angle advantageously is within the range described heretofore with respect to the parabolic surface 107 (FIGURE 2). A comparatively straight segment 188 of the member 185 merges with the parabolic segment and is followed by a second straight segment 189. This latter segment is disposed adjacent the lowermost portion of the member 185, this portion being in the shape of a hollow cylinder.

To insure that the ambient fluid drawn into the Coanda slot 181 is derived from the upper portion of the apparatus, there is provided a cup-shaped shield 190 which is axially disposed along the flow axis 130. The rim of the shield 190 is affixed to a flat ring 192 rigidly secured to the lower conduit 152, while the center shield portion substantially closes the lower end of the elongated tube 133. The ring 192 also serves as support for the inner portion of the lip member 185.

An axial opening **195** of a diameter less than that of the tube **133** is provided in the shield **190**. The opening **195** accommodates a comparatively short vertically oriented tube **197** which is closed at its lower end. As best shown in FIGURE 6, four elongated pipes **200**, **201**, **202** and **203** extend radially from the tube **197** at right angles to each other and are connected at their outer ends to the inner surface of the member **185** by brackets **205**. Each of these pipes includes a series of longitudinally spaced openings **206** therein which communicate with the space defined by the inner surface of the member **185**. The openings **206** are supplied with fluid under pressure from the elongated tube **133** and the tube **197**. The fluid flowing through the openings prevents any substantial low pressure area in the space which might otherwise adversely affect the lift of the apparatus. In addition, the pipes provide structural support for the member **185**.

An upstanding shroud **210** surrounds the lip member **185** in spaced coaxial relationship therewith. The shroud **210** is in the form of a hollow cylinder and defines a duct **211** between its inner surface and the outer surface of the member **185**. The shroud **210** and the member **185** are fixedly secured together by four comparatively thin brackets **212** (FIGURE 6) which extend in an outward direction from the portion **189** of the member **185**.

The fluid discharged from the third stage Coanda slot **181** emerges along the horizontal plane **175** and is directed downwardly over the parabolic lip **187** of the member **185** in accordance with the Coanda effect. This fluid entrains a substantial quantity of ambient air from the upper portion of the device. The entrained air is drawn into the duct **211** and is discharged from the lower portion thereof in the direction of the flow axis **130** to form a zone of increased pressure beneath the device. The resulting pressure differential produces an axially directed lifting force of substantial magnitude.

Because of the pressurized fluid flowing from the pipes **202** into the partially enclosed space defined by the inner surface of the lip member **185**, the pressure in this space is increased to a pressure which is at least equal to and preferably slightly greater than atmospheric pressure. With this arrangement, any pressure reduction in the space occasioned as a result of the entrainment of ambient air therein by the tertiary fluid flow through the duct **211** is maintained at a minimum.

In each of the illustrated embodiments of the invention, the propulsion force is produced as a result of a first stage Coanda slot which entrains a primary flow of ambient air, a pair of second stage Coanda slots which entrains a secondary ambient flow and a third stage Coanda slot which entrains a tertiary flow. The various flows are combined and are directed axially with respect to the device to produce the desired lift while avoiding any necessity for the use of moving parts or highly combustible fluids.

In other good arrangements, particularly for comparatively small aircraft having correspondingly lower propulsion requirements, the first stage Coanda slot is eliminated. In these latter situations, the ambient air entrained by the fluid flowing through the pair of Coanda slots **64** and **65** of FIGURE 2, for example, comprises the primary flow, while that entrained by the fluid flowing through the Coanda slot **181** comprises the secondary flow.

Although the invention has been shown and described as having particular application in the propulsion of aircraft by means of gaseous fluid acting on ambient air, it is not, in its broadest aspects, restricted to such application. Thus, the invention is broadly applicable to the movement of a vehicle through a wide variety of ambient mediums through the use of various pressurized fluids. As an illustration, in some cases the apparatus is useful as a propulsion device for underwater vehicles, for example, in which the ambient liquid is entrained by pressurized steam, air, water, or other fluid, to produce the desired propulsion force. Other uses for the apparatus will suggest themselves to those skilled in the art upon a perusal of the foregoing description.

The terms and expressions which have been employed are used as terms of description and not of limitation, and there is no intention in the use of such terms and expressions of excluding any equivalents of the features shown and described or portions thereof, but it is recognized that various modifications are possible within the scope of the invention claimed.

What is claimed is:

1. In propulsion apparatus, in combination, first and second fluid conductor means symmetrically disposed about an axis of flow in spaced-apart relationship with each other, said conductor means respectively including first and second Coanda slots and the facing portions of said conductor means defining a third Coanda slot, a supply of fluid under pressure, means for directing fluid from said supply to said conductor means and through said first and second Coanda slots, the fluid passing through said first and second Coanda slots inducing a primary flow of ambient fluid into said third Coanda slot, and lip means on the downstream side of said third Coanda slot and having a curved surface in juxtaposition therewith, the fluid flowing through said third Coanda slot inducing a secondary flow of ambient fluid over said curved surface and in the direction of said axis, to thereby produce an axially directed propulsion force.

2. In propulsion apparatus, in combination, first and second fluid conductor means of generally circular configuration in spaced-apart coaxial relationship with each other, said conductor means respectively including first and second Coanda slots and the facing portions of said conductor means defining a third Coanda slot, a supply of fluid under pressure, means for directing fluid from said supply to said conductor means and simultaneously through said first and second Coanda slots, the fluid passing through said first and second Coanda slots inducing a primary flow of ambient fluid into said third Coanda slot, and lip means symmetrically disposed about the axis of said conductor means and having a curved surface in juxtaposition with said third Coanda slot, the fluid flowing through said third Coanda slot inducing a secondary flow of ambient fluid over said curved surface and in the direction of said axis, to thereby produce an axially directed propulsion force.

3. In propulsion apparatus, in combination, first and second fluid conductor means symmetrically disposed about an axis of flow in spaced-apart relationship with each other, each of said conductor means having an annular surface thereon, said conductor means respectively including first and second Coanda slots cooperating with the corresponding surface and the facing portions of said conductor means defining a third Coanda slot, a supply of fluid under pressure, means for directing fluid from said supply to said conductor means and through said first and second Coanda slots, the fluid passing through said first and second Coanda slots inducing a primary flow of ambient fluid into said third Coanda slot, and lip means on the downstream side of said third Coanda slot and having a curved surface in juxtaposition therewith, said curved surface being defined by a parabolic segment revolved about the axis of flow, the fluid flowing through said third Coanda slot inducing a secondary flow of ambient fluid over said curved surface and in the direction of said axis, to thereby produce an axially directed propulsion force.

4. In propulsion apparatus, in combination, first and second fluid conductor means of annular configuration in spaced-apart coaxial relationship with each other, each of said conductor means having an annular surface thereon, said conductor means respectively including first and second Coanda slots cooperating with the corresponding surface and the facing portions of said conductor means defining a third Coanda slot, a supply of fluid under pressure, means for directing fluid from said

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supply to said conductor means and through said first and second Coanda slots, the fluid passing through said first and second Coanda slots inducing a primary flow of ambient fluid into said third Coanda slot, and a lip member of circular cross section coaxially supported with respect to said conductor means and having a curved surface in juxtaposition with said third Coanda slot, said curved surface being defined by a parabolic segment revolved about the axis of said member, the fluid flowing through said third Coanda slot inducing a secondary flow of ambient fluid over said curved surface and in the direction of said axis, to thereby produce an axially directed propulsion force.

5. In discoid propulsion apparatus, in combination, first and second annular conduits in spaced-apart coaxial relationship with each other, each of said conduits having a surface of parabolic configuration thereon, said conduits respectively including first and second Coanda slots cooperating with the corresponding surface and the facing portions of said conduits defining a third Coanda slot, a supply of fluid under pressure, means for directing fluid from said supply to said conduits and through said first and second Coanda slots, the fluid passing through said first and second Coanda slots inducing a primary flow of ambient fluid into said third Coanda slot, and an annular member coaxially supported with respect to said conduits and having a curved surface in juxtaposition with said third Coanda slot, said curved surface being defined by a parabolic segment revolved about the axis of said member, the fluid flowing through said third Coanda slot inducing a secondary flow of ambient fluid over said curved surface and in the direction of said axis, to thereby produce an axially directed propulsion force.

6. In propulsion apparatus, in combination, first and second conduits symmetrically disposed about an axis of flow in spaced-apart relationship with each other, each of said conduits having an annular surface thereon coaxial with said flow axis, said conduits respectively including first and second annular Coanda slots cooperating with the corresponding surface and the facing portions of said conduits defining a third annular Coanda slot, a supply of fluid under pressure, means for directing fluid from said supply to said conduits and simultaneously through said first and second Coanda slots, the fluid passing through said first and second Coanda slots inducing a primary flow of ambient fluid into said third Coanda slot, and an annular member on the downstream side of said third Coanda slot and having a parabolic surface the axis of which forms an actue angle with the axis of flow, the fluid flowing through said third Coanda slot inducing a secondary flow of ambient fluid over said parabolic surface and in the direction of said axis, to thereby produce an axially directed propulsion force.

7. Apparatus of the character set forth in claim 6, in which said acute angle is within the range of from about 1[illegible] degrees to about 45 degrees.

8. In propulsion apparatus of generally discoid configuration, in combination, first and second annular conduits in spaced-apart relationship with each other, each of said conduits being symmetrically disposed about an axis of flow and having a parabolic surface thereon, said conduits respectively including first and second annular Coanda slots cooperating with the corresponding surface and the facing portions of said conduits defining a third annular Coanda slot, a supply of fluid under pressure, means for directing fluid from said supply to said conduits and through said first and second Coanda slots, the fluid passing through said first and second Coanda slots emerging in a direction parallel to said axis of flow and inducing a primary flow of ambient fluid in a direction transverse to said axis, the fluid from said first and second Coanda slots and said ambient fluid being directed into said third Coanda slot, and a lip member of circular cross-section coaxially supported with respect to said conduits and having a curved surface in juxtaposition with said third Coanda slot, said curved surface being defined by a parabolic segment revolved about said flow axis, the fluid flowing through said third Coanda slot inducing a secondary flow of ambient fluid over said curved surface and in the direction of said axis, to thereby produce an axially directed propulsion force.

9. In propulsion apparatus of generally discoid configuration, in combination, first and second annular conduits in spaced-apart coaxial relationship with each other, each of said conduits being symmetrically disposed about an axis of flow and having a parabolic external surface thereon, said conduits respectively including first and second annular Coanda slots cooperating with the corresponding surface and the facing portions of said conduits defining a third annular Coanda slot, a supply of fluid under pressure, means for directing fluid from said supply to said conduits and simultaneously through said first and second Coanda slots, the fluid passing through said first and second Coanda slots emerging in a direction parallel to said axis of flow and inducing a primary flow of ambient fluid inwardly toward said axis in a direction transverse thereto from the periphery of the apparatus, the fluid from said first and second Coanda slots and said ambient fluid being directed into said third Coanda slot, and an annular lip member coaxially supported with respect to said conduits and having a curved surface in juxtaposition with the inner portion of said third Coanda slot, said curved surface being defined by a parabolic segment revolved about said flow axis, the fluid flowing through said third Coanda slot inducing a secondary flow of ambient fluid over said curved surface and in the direction of said axis, to thereby produce an axially directed propulsion force.

10. In propulsion apparatus of generally discoid configuration, in combination, first and second annular conduits in spaced-apart coaxial relationship with each other, each of said conduits being symmetrically disposed about an axis of flow and having a parabolic internal surface thereon, said conduits respectively including first and second annular Coanda slots cooperating with the corresponding surface and the facing portions of said conduits defining a third annular Coanda slot, a supply of fluid under pressure, means for directing fluid from said supply to said conduits and simultaneously through said first and second Coanda slots, the fluid passing through said first and second Coanda slots emerging in a direction parallel to said axis of flow and inducing a primary flow of ambient fluid outwardly away from said axis in a direction transverse thereto, the fluid from said first and second Coanda slots and said ambient fluid being directed into said third Coanda slot, and a lip member of circular cross-section coaxially supported with respect to said conduits and having a curved surface in juxtaposition with the outer portion of said third Coanda slot, said curved surface being defined by a parabolic segment revolved about said flow axis, the fluid flowing through said third Coanda slot inducing a secondary flow of ambient fluid over said curved surface and in the direction of said axis, to thereby produce an axially directed propulsion force.

11. Propulsion apparatus comprising, in combination, first and second conduits of annular configuration in spaced-apart coaxial relationship with each other, each of said conduits having a curved surface thereon, said conduits respectively including first and second Coanda slots cooperating with the corresponding surface and the facing portions of said conduits defining a third Coanda slot, a supply of fluid under pressure, fluid conductor means connected between said supply and said conduits, said conductor means including an additional Coanda slot supplied with said pressurized fluid and communicating with the atmosphere, the fluid supplied to said additional Coanda slot inducing a primary flow of ambient fluid into said conduits and through said first and second Coanda slots, the fluid passing through said first and second Coanda slots inducing a secondary flow of ambient fluid into said third Coanda slot, and a lip member of circular cross-section coaxially supported with respect to said conduits

and having a curved surface adjacent the discharge side of said third Coanda slot, said curved surface being defined by a parabolic segment revolved about the axis of said member, the fluid flowing through said third Coanda slot inducing a tertiary flow of ambient fluid over said curved surface and in the direction of said axis, to thereby produce an axially directed propulsion force.

12. A discoid propulsion device for producing an axially directed propulsion force comprising, in combination, first and second annular conduits in spaced-apart coaxial relationship with each other, each of said conduits having an annular surface of parabolic configuration, said conduits respectively including first and second Coanda slots cooperating with the corresponding surface and the facing portions of said conduits defining a third Coanda slot, a supply of fluid under pressure, fluid conductor means connected between said supply and said conduits, said conductor means including an additional Coanda slot having a parabolic discharge lip, said additional Coanda slot being supplied with said pressurized fluid and communicating with the atmosphere to induce a primary flow of ambient fluid over said discharge lip, into said conduits and through said first and second Coanda slots, the fluid passing through said first and second Coanda slots inducing a secondary flow of ambient fluid into said third Coanda slot, and a lip member of circular cross-section coaxially supported with respect to said conduits and having a curved surface adjacent the discharge side of said third Coanda slot, said curved surface being defined by a parabolic segment revolved about the axis of said lip member, the fluid flowing through said third Coanda slot inducing a tertiary flow of ambient fluid over said curved surface and in the direction of said axis, to thereby produce an axially directed propulsion force.

13. Propulsion apparatus comprising, in combination, first and second annular conduits symmetrically disposed about an axis of flow in spaced-apart coaxial relationship with each other, each of said conduits having an annular parabolic surface thereon, said conduits respectively including first and second annular Coanda slots cooperating with the corresponding surface and the facing portions of said conduits defining a third annular Coanda slot, a supply of fluid under pressure, fluid conductor means connected between said supply and said conduits, said conductor means including an additional annular Coanda slot having a first parabolic discharge lip, said additional Coanda slot being supplied with said pressurized fluid and communicating with the atomsphere to induce a primary flow of ambient fluid over said first discharge lip, said first lip having a first parabolic axis which meets the direction of primary flow at a first acute angle, said conductor means directing said pressurized fluid and said primary flow from said first lip into said conduit and through said first and second Coanda slots, the fluid passing through said first and second Coanda slots inducing a secondary flow of ambient fluid into said third Coanda slot, the annular surface of each of said conduits having a second parabolic axis which meets the direction of secondary flow at a second acute angle, and a lip member of circular cross-section coaxially supported with respect to said conduits and having a second parabolic discharge lip adjacent said third Coanda slot, said second lip being defined by a parabolic segment revolved about said flow axis, the fluid flowing through said third Coanda slot inducing a tertiary flow of ambient fluid over said second lip, said second lip having a third parabolic axis which meets the direction of tertiary flow at a third acute angle, to thereby produce an axially directed propulsion force.

14. Lift producing apparatus of generally discoid configuration comprising, in combination, first and second annular conduits in spaced-apart coaxial relationship with each other, each of said conduits having an annular parabolic surface thereon, said conduits respectively including first and second annular Coanda solts cooperating with the corresponding surface and the facing portions of said conduits defining a third annular Coanda slot, a supply of fluid under pressure, fluid conductor means connected between said supply and said conduits, said conductor means including an additional annular Coanda slot having a first parabolic discharge lip, said additional Coanda slot being supplied with said pressurized fluid and communicating with the atmosphere to induce a primary flow of ambient fluid over said first discharge lip, said conductor means directing said pressurized fluid and said primary flow from said first lip into said conduits and simultaneously through said first and second Coanda slots, the fluid passing through said first and second Coanda slots inducing a secondary flow of ambient fluid into said third Coanda slot, and a lip member of circular cross-section coaxially supported with respect to said conduits and having a second parabolic discharge lip adjacent said third Coanda slot, said second lip being defined by a parabolic segment revolved about the axis of said member, the fluid flowing through said third Coanda slot inducing a tertiary flow of ambient fluid over said second lip and in the direction of said axis, to thereby produce a propulsion force having a axially directed lift component.

15. Apparatus of the character set forth in claim 14, in which said fluid conductor means includes a plurality of radially extending pipes interconnecting said additional Coanda slot with said first and second conduits, each of said pipes having airfoil means of insulating material on its exposed outer surface.

16. A discoid propulsion device for producing an axially directed propulsion force comprising, in combination, first and second conduits of annular configuration symmetrically disposed about an axis of flow in spaced-apart coaxial relationship with each other, each of said conduits having an annular parabolic surface thereon, said conduits respectively including first and second annular Coanda slots cooperating with the corresponding surface and the facing portions of said conduits defining a third annular Coanda slot, a supply of fluid under pressure, fluid conductor means connected between said supply and said conduits, said conductor means including an additional annular Coanda slot in coaxial relationship with said flow axis and having a first parabolic discharge lip, said additional Coanda slot being supplied with said pressurized fluid and communicating with the atmosphere to induce a primary flow of ambient fluid along said flow axis and over said first discharge lip, said first lip having a first parabolic axis which meets the direction of primary flow at a first acute angle, said conductor means directing said pressurized fluid and said primary flow from adjacent said first lip into said conduits and simultaneously through said first and second Coanda slots in a direction parallel to said flow axis, the fluid passing through said first and second Coanda slots inducing a secondary flow of ambient fluid into said third Coanda slot from a direction transverse to said flow axis, the annular surface of each of said conduits having a second parabolic axis which meets the direction of secondary flow at a second acute angle, and a lip member of circular cross-section coaxially supported with respect to said conduits and having a second parabolic discharge lip adjacent said third Coanda slot, said second lip being defined by a parabolic segment revolved about said flow axis, the fluid flowing through said third Coanda slot inducing a tertiary flow of ambient fluid over said second lip and in the direction of said axis, said second lip having a third parabolic axis which meets the direction of tertiary flow at a third acute angle, to thereby produce an axially directed propulsion force.

17. Apparatus of the character set forth in claim 16, in which said lip member is shaped to define a partially enclosed space adjacent said annular conduits, means supported in said space and supplied with fluid under pressure for increasing the pressure therein, and means for preventing the entrainment of ambient fluid from said space into said third Coanda slot.

18. A discoid propulsion device for producing an

axially directed propulsion force comprising, in combination, first and second conduits of annular configuration symmetrically disposed about an axis of flow in spaced-apart coaxial relationship with each other, each of said conduits having an annular internal parabolic surface thereon, said conduits respectively including first and second annular Coanda slots cooperating with the corresponding surface and the facing portions of said conduits defining a third annular Coanda slot, a supply of fluid under pressure, fluid conductor means connected between said supply and said conduits, said conductor means including an additional annular Coanda slot in coaxial relationship with said flow axis and having a first parabolic discharge lip, said additional Coanda slot being supplied with said pressurized fluid and communicating with the atmosphere to induce a primary flow of ambient fluid along said flow axis and over said first discharge lip, said first lip having a first parabolic axis which meets the direction of primary flow at a first acute angle, said conductor means including a first group of radially extending fluid conductors connected to said first conduit and a second group of radially extending fluid conductors connected to said second conduit for directing said pressurized fluid and said primary flow from adjacent said first lip into the corresponding conduits and simultaneously through said first and second Coanda slots in a direction parallel to said flow axis, the fluid passing through said first and second Coanda slots inducing a secondary flow of ambient fluid outwardly away from said flow axis in a direction transverse thereto and into said third Coanda slot, the annular surface of each of said conduits having a second parabolic axis which meets the direction of secondary flow at a second acute angle, and a lip member of circular cross-section coaxially supported with respect to said conduits and having a second parabolic discharge lip adjacent the outer portion of said third Coanda slot, said second lip being defined by a parabolic segment revolved about said flow axis, the fluid flowing through said third Coanda slot inducing a tertiary flow of ambient fluid over said second lip and in the direction of said axis, said second lip having a third parabolic axis which meets the direction of tertiary flow at a third acute angle, to thereby produce an axially directed propulsion force.

19. Apparatus of the character set forth in claim 18, which comprises shroud means symmetrically supported around said lip member in spaced relationship therewith, said shroud means and said lip member forming a duct for the passage of said tertiary fluid flow.

20. A discoid propulsion device for producing an axially directed propulsion force comprising, in combination, first and second conduits of annular configuration symmetrically disposed about an axis of flow in spaced-apart coaxial relationship with each other, each of said conduits having an annular external parabolic surface thereon, said conduits respectively including first and second annular Coanda slots cooperating with the corresponding surface and the facing portions of said conduits defining a third annular Coanda slot, a supply of fluid under pressure, fluid conductor means connected between said supply and said conduits, said conductor means including an additional annular Coanda slot in coaxial relationship with said flow axis and having a first parabolic discharge lip, said additional Coanda slot being supplied with said pressurized fluid and communicating with the atmosphere to induce a primary flow of ambient fluid along said flow axis and over said first discharge lip, said first lip having a first parabolic axis which meets the direction of primary flow at a first acute angle, said conductor means including a plurality of radially extending fluid conductors for directing said pressurized fluid and said primary flow from adjacent said first lip into said conduits and simultaneously through said first and second Coanda slots in a direction parallel to said flow axis, each of said fluid conductors being connected to said first conduit and to said second conduit, the fluid passing through said first and second Coanda slots inducing a secondary flow of ambient fluid inwardly toward said flow axis in a direction transverse thereto and into said third Coanda slot, the annular surface of each of said conduits having a second parabolic axis which meets the direction of secondary flow at a second acute angle, and a lip member of circular cross-section coaxially supported with respect to said conduits and having a second parabolic discharge lip adjacent the inner portion of said third Coanda slot, said second lip being defined by a parabolic segment revolved about said flow axis, the fluid flowing through said third Coanda slot inducing a tertiary flow of ambient fluid over said second lip and in the direction of said axis, said second lip having a third parabolic axis which meets the direction of tertiary flow at a third acute angle, to thereby produce an axially directed propulsion force.

No references cited.

MARK NEWMAN, *Primary Examiner.*

C. R. CROYLE, *Assistant Examiner.*

UNITED STATES PATENT OFFICE
CERTIFICATE OF CORRECTION

Patent No. 3,261,162 60 July 1 , 1966

Henri Coanda

It is hereby certified that error appears in the above numbered patent requiring correction and that the said Letters Patent should read as corrected below.

Column 2, 8, for "from" read -- form --; column 3, line read -- aircraft --; line 71, for "removed"

Fluid Diode

Inventor's creations still prove electrifying

■ Science:

Contemporary engineers find a new use for an old, cleverly designed pump.

By The Associated Press

SEATTLE — When two University of Washington engineers and their students set out to build a tiny pump for use in medicine, electronics and other arenas, they got some insight from a 75-year-old patent held by inventor Nikola Tesla.

The work of Tesla, who was once a household name, had a profound effect on the modern world. He invented fluorescent light bulbs, held the first patent on radio and has his name on more than 700 U.S. patents. Most of our electrical power system — based on alternating current — was Tesla's creation.

"His impact on the 20th century was phenomenal," said Martin Afromowitz, a UW professor of electrical engineering.

Yet many of his ideas, including a pump design known as the Tesla valve, have languished in obscurity.

"I was surprised at how little application this (pump design) had had," said Fred Forster, a UW professor of mechanical engineering.

Tesla had built a gas engine pump without moving parts. The pump featured two complicated series of loops for the intake and outlet valves designed to favor flow in one direction over the other.

This "leaky valve" approach to pumping is not very efficient in terms of overall flow rate, Forster said, but it has the advantage of being simple in construction and highly reliable.

Forster, Afromowitz and graduate students Nigel Sharma and Ron Bardell studied Tesla's 1920 patent and created a variety of tiny Tesla-style pumps by etching silicon wafers in the same way computer chips are created.

"It looks kind of bizarre," Afromowitz said. "We had no idea if this would work."

It did, but only after the UW team adapted Tesla's design to fluid behavior at the micro level. Some $200,000 in private and public funding was made available to the UW engineering team now working to perfect this micropump, which has no moving parts.

The intake and outlet valves, each a series of pretzel-like loops on each side of the pump chamber, are about the width of a human hair. The central pump chamber, also silicon, is maybe a quarter-inch in diameter and one-tenth of a millimeter thick.

A small electric charge applied hundreds of times per second to the pump chamber makes it flex, forcing the fluid to move. Because of the design of the valves, the fluid tends to move in one direction.

"We expect miniaturized fluid systems to be a major breakthrough in new technologies," Forster said.

If so, it's another breakthrough credited to a man someone once called "the greatest inventor the world has ever known."

Tesla, a Serb born in Croatia in 1856, came to the United States in 1884 to work in Thomas Edison's lab. The two had a falling out.

He would go on to create the alternating current induction motor, which made Westinghouse an industrial giant and which still represents about 80 percent of all electrical power in use today.

Tesla died in 1943, a semirecluse and in debt.

and from the conservation of angular momentum

$$u_{00} = (r_i/r_0)u_{\theta i} \tag{2.34}$$

The substitution of Eqs. (2.33) and (2.34) into Eq. (2.32) yields

$$p_1 - p_0 = (\rho/2)[u_{ri}^2 + u_{\theta i}^2][(r_i^2/r_0^2) - 1] \tag{2.35}$$

Since u_{ri} is much smaller than $u_{\theta i}$ we may select the reference velocity $U = u_{\theta i}$ and thus Eq. (2.35) becomes

$$(K_A)_{12} = \frac{p_1 - p_0}{\rho U^2/2} = \left(\frac{r_i}{r_0}\right)^2 - 1 \tag{2.36}$$

From Eqs. (2.31) and (2.36) the total reverse flow coefficient for the vortex diode becomes

$$K_{LR} = (r_i/r_0)^2 \tag{2.37}$$

Equation (2.37) indicates that we can achieve large reverse flow coefficients by simply making the radius ratio large. However, because of viscosity the loss coefficients will be much less than predicted by Eq. (2.37).

In the forward flow direction the loss coefficient is essentially the result of two sudden enlargements. Thus we may approximate the forward loss coefficient as $K_{LF} = 2$.

The performance ratio of the vortex diode can now be estimated. For example, if we assume incompressible flow and a vortex chamber with a radius ratio of 5, the reverse loss coefficient is 25 and the forward loss coefficient is 2. Thus the performance ratio is 12.5. This estimate agrees remarkably well with test data [30]. Vortex diodes with larger radius ratios and therefore higher performance ratios ($D_i = 43$) have been reported [29]. These devices produced less diodicity than the present theory predicts. However the circuits contained other diode components in addition to the vortex chamber so that a direct comparison is not possible. We will discuss the vortex diode as a special case of the vortex triode in Chapter 8.

The cascade diode is another fluid diode based on rotational flow in one direction and separated type flow in the other [30, 31]. Figure 2.20 shows a schematic drawing of a cascade diode that has four cascades, each with six blades, mounted inside a tube. Forward flow (left to right on Fig. 2.20) passes in the axial direction between the cascades. Reverse flow (right to left), on the other hand, is deflected by the blades so that the flow rotates from one cascade to another. For reverse flow the first blade in each cascade makes a 45-degree angle with the axis. The following blades have progressively larger angles until they reach 90 degrees. These latter blades are the first blades contacted in the forward flow. Since they are set at 90 degrees they impart no spin to the fluid. The diode shown in Fig. 2.20 is a straight cascade because the blades are mounted directly behind each other. However, the rotation of the fluid is enhanced when each cascade follows a helical pattern. Although forward resistance is also increased by this means, there is a larger increase in reverse resistance and diodicity increases.

Design theory of Fluidic Components
Joseph M. Kirshner & Silas Katz

The performance of the cascade diode depends mainly on the number of cascades, the number of blades per cascade, and the helix angle. Performance ratios up to 181 have been obtained for a four-cascade, 53 blades per cascade, diode with a helix angle of 90 degrees [31]. The calculations of Jacobs and Baker [31] indicate that improvement of the performance to about 350 is possible if the number of blades per cascade is increased.

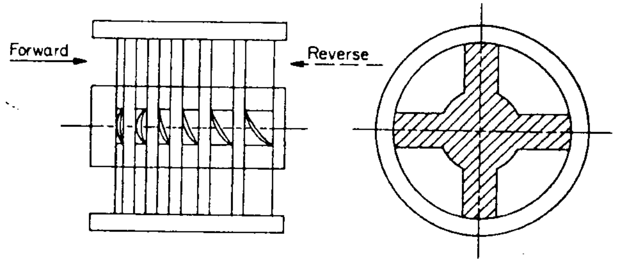

Fig. 2.20 Cascade diode.

A disadvantage of the cascade diode is that it is geometrically complex and, therefore, expensive to fabricate. In addition there is the possibility of deterioration of the performance through mechanical damage to the blades.

2.7.2 Momentum Interaction Diodes

"fluid diode"

The scroll diode [30] and the Tesla diode [19, 32] use momentum interaction to achieve reverse flow resistance.

Figure 2.21a shows the configuration of the scroll diode. This diode is essentially a converging nozzle with a sudden enlargement that has an axial annular cup. In the reverse flow direction fluid separates at the nozzle throat and enters the annular cup. The cup directs the fluid back toward the oncoming flow. Finally the flow turns again to exit through the annular passage between the cup and the sudden enlargement. In the forward direction the fluid does not enter the cup. The fluid flows through the throat and then into a diffuser section to minimize forward losses.

Measurements on scroll diodes of various geometries have yielded performance ratios in the range from 3 to 7 [30]. With careful redesign the ratio may increase to about 10. This, however, seems to be the upper limit of performance for the scroll diode.

The Tesla diode (Fig. 2.21b) is a series of connected branches and flow loops. In the reverse flow direction fluid enters the branches at a small angle and loops

Academic Press Inc. NY 75

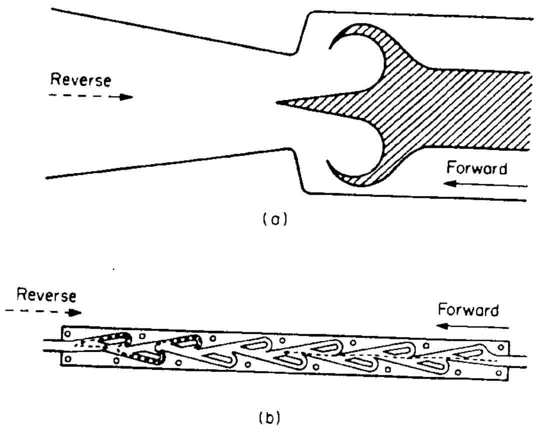

Fig. 2.21 Momentum interaction diodes: (a) scroll diode, (b) Tesla diode.

around to oppose the main flow. Forward flow is predominantly axial in direction and incurs an enlargement loss when it passes over the branching sections.

Figure 2.22a shows a schematic model of a single stage Tesla diode [32]. In reverse flow, fluid passing section 0 divides between the in-line section 1 and the branch section 2. The branch flow then loops around and is directed at the in-line flow. The two flows then recombine to enter section 3.

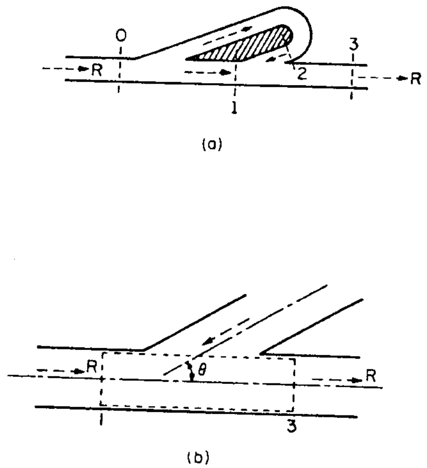

Fig. 2.22 Tesla diode model for reverse flow: (a) single-stage Tesla diode, (b) momentum interaction model.

The control volume shown in Fig. 2.22b provides a method of determining the reverse loss coefficient [32]. Application of the momentum equation between sections 1 and 3 leads to the expression

$$p_1 - p_3 = \rho \bar{U}_3^2 - \rho \bar{U}_1^2 + (A_2/A_3)\bar{U}_2^2 \cos\theta \qquad (2.38)$$

where the areas are A_1, A_2, and A_3 ($A_1 = A_3$) and θ is the angle that the return loop makes with the in-line direction. From Eq. (2.38), after some rearrangement, the reverse loss coefficient becomes

$$K_{LR} = 1 - \bar{U}_1^2/\bar{U}_3^2 + 2(A_2/A_3)(\bar{U}_2^2/\bar{U}_3^2)\cos\theta \qquad (2.39)$$

If we now assume that the static and total pressures are equal at sections 1 and 2, then the velocities must also be equal ($\bar{U}_1 = \bar{U}_2$). With this relation and the continuity equation, Eq. (2.39) reduces to

$$K_{LR} = \frac{(A_2/A_3)^2 + 2(A_2/A_3)(1 + \cos\theta)}{(1 + A_2/A_3)^2} \qquad (2.40)$$

The maximum reverse loss coefficient occurs when $\theta = 0$ degrees and $A_2/A_3 = 2$ and has a magnitude of 1.33. Of course it is not possible practically to make the angle zero. Nevertheless this value is useful as an indication of the maximum possible magnitude. In forward flow this configuration is a type of sudden enlargement, and therefore we would expect the forward total loss coefficient to fall between 0 and 1 (Fig. 2.15). Test data on typical geometries of single-stage Tesla diode [32] result in reverse coefficients of about 1 and forward coefficients of about 0.25. Thus the performance ratio of the diode is in the vicinity of 4. Series or parallel connection of a number of single-stage Tesla diodes should not alter performance to any large extent since both forward and reverse resistances change in the same proportion.

2.8 RESISTANCE NETWORKS

2.8.1 Branch Resistance

A fluidic resistive circuit that occurs frequently is the branch or junction. Figure 2.23 shows a schematic drawing of the equal-area TEE branch. For dividing flow (Fig. 2.23a), fluid enters the in-line section 1 and leaves through in-line section 2 and branch-line section 3. For mixing flow (Fig. 2.23b), fluid flows in the opposite direction in the branch line. Thus fluid enters at sections 1 and 3, and leaves at section 2. The flow pattern at the junction is different in mixing and dividing flows. As a result the loss coefficients for the two cases also differ.

Let us consider the dividing flow condition as a variable area passage with entrance at section 1 and exits at sections 2 and 3. If we assume that the sections are far enough from the junction to insure uniform velocity and pressure distributions the energy equation [Eq. (2.8)] for the branch is

$$p_1\bar{U}_1 = p_2\bar{U}_2 + p_3\bar{U}_3 + (K_L)_{12}(\rho\bar{U}_1^2/2)\bar{U}_2 + (K_L)_{13}(\rho\bar{U}_1^2/2)\bar{U}_3 \qquad (2.41)$$

UNITED STATES PATENT OFFICE.

NIKOLA TESLA, OF NEW YORK, N. Y.

VALVULAR CONDUIT.

1,329,559. Specification of Letters Patent. **Patented Feb. 3, 1920.**

Application filed February 21, 1916, Serial No. 79,703. Renewed July 8, 1919. Serial No. 309,482.

To all whom it may concern:

Be it known that I, Nikola Tesla, a citizen of the United States, residing at New York, in the county and State of New York, have invented certain new and useful Improvements in Valvular Conduits, of which the following is a full, clear, and exact description.

In most of the machinery universally employed for the development, transmission and transformation of mechanical energy, fluid impulses are made to pass, more or less freely, through suitable channels or conduits in one direction while their return is effectively checked or entirely prevented. This function is generally performed by devices designated as valves, comprising carefully fitted members the precise relative movements of which are essential to the efficient and reliable operation of the apparatus. The necessity of, and absolute dependence on these, limits the machine in many respects, detracting from its practical value and adding greatly to its cost of manufacture and maintenance. As a rule the valve is a delicate contrivance, very liable to wear and get out of order and thereby imperil ponderous, complex and costly mechanism and, moreover, it fails to meet the requirements when the impulses are extremely sudden or rapid in succession and the fluid is highly heated or corrosive.

Though these and other correlated facts were known to the very earliest pioneers in the science and art of mechanics, no remedy has as yet been found or proposed to date so far as I am aware, and I believe that I am the first to discover or invent any means, which permit the performance of the above function without the use of moving parts, and which it is the object of this application to describe.

Briefly expressed, the advance I have achieved consists in the employment of a peculiar channel or conduit characterized by valvular action.

The invention can be embodied in many constructions greatly varied in detail, but for the explanation of the underlying principle it may be broadly stated that the interior of the conduit is provided with enlargements, recesses, projections, baffles or buckets which, while offering virtually no resistance to the passage of the fluid in one direction, other than surface friction, constitute an almost impassable barrier to its flow in the opposite sense by reason of the more or less sudden expansions, contractions, deflections, reversals of direction, stops and starts and attendant rapidly succeeding transformations of the pressure and velocity energies.

For the full and complete disclosure of the device and of its mode of action reference is made to the accompanying drawings in which—

Figure 1 is a horizontal projection of such a valvular conduit with the top plate removed.

Fig. 2 is side view of the same in elevation.

Fig. 3 is a diagram illustrative of the application of the device to a fluid propelling machine such as, a reciprocating pump or compressor, and

Fig. 4 is a plan showing the manner in which the invention is, or may be used, to operate a fluid propelled rotary engine or turbine.

Referring to Fig. 1, 1 is a casing of metal or other suitable material which may be cast, milled or pressed from sheet in the desired form. From its side-walls extend alternatively projections terminating in buckets 2 which, to facilitate manufacture are congruent and spaced at equal distances, but need not be. In addition to these there are independent partitions 3 which are deemed of advantage and the purpose of which will be made clear. Nipples 4 and 5, one at each end, are provided for pipe connection. The bottom is solid and the upper or open side is closed by a fitting plate 6 as shown in Fig. 2. When desired any number of such pieces may be joined in series, thus making up a valvular conduit of such length as the circumstances may require.

In elucidation of the mode of operation let it be assumed that the medium under pressure be admitted at 5. Evidently, its approximate path will be as indicated by the dotted line 7, which is nearly straight, that is to say, if the channel be of adequate cross-section, the fluid will encounter a very small resistance and pass through freely and undisturbed, at least to a degree. Not so if the entrance be at the opposite end 4. In this case the flow will not be smooth and continuous, but intermittent, the fluid being quickly deflected and reversed in direction, set in whirling motion, brought to rest and again accelerated, these processes following one another in rapid succession. The partitions 3 serve to direct the stream upon the buckets and to intensify the actions causing violent surges and eddies which interfere very materially with the flow through the conduit. It will be readily observed that the resistance offered to the passage of the medium will be considerable even if it be under constant pressure, but the impediments will be of full effect only when it is supplied in pulses and, more especially, when the same are extremely sudden and of high frequency. In order to bring the fluid masses to rest and to high velocity in short intervals of time energy must be furnished at a rate which is unattainable, the result being that the impulse cannot penetrate very far before it subsides and gives rise to movement in the opposite direction. The device not only acts as a hinderment to the bodily return of particles but also, in a measure, as a check to the propagation of a disturbance through the medium. Its efficacy is chiefly determined; first, by the magnitude of the ratio of the two resistances offered to disturbed and to undisturbed flow, respectively, in the directions from 4 to 5 and from 5 to 4, in each individual element of the conduit; second, by the number of complete cycles of action taking place in a given length of the valvular channel and, third, by the character of the impulses themselves. A fair idea may be gained from simple theoretical considerations.

Examining more closely the mode of operation it will be seen that, in passing from one to the next bucket in the direction of disturbed flow, the fluid undergoes two complete reversals or deflections through 180 degrees while it suffers only two small deviations from about 10 to 20 degrees when moving in the opposite sense. In each case the loss of head will be proportionate to a hydraulic coefficient dependent on the angle of deflection from which it follows that, for the same velocity, the ratio of the two resistances will be as that of the two coefficients. The theoretical value of this ratio may be 200 or more, but must be taken as appreciably less although the surface friction too is greater in the direction of disturbed flow. In order to keep it as large as possible, sharp bends should be avoided, for these will add to both resistances and reduce the efficiency. Whenever practicable, the piece should be straight; the next best is the circular form.

That the peculiar function of such a conduit is enhanced by increasing the number of buckets or elements and, consequently, cyclic processes in a given length is an obvious conclusion, but there is no direct proportionality because the successive actions diminish in intensity. Definite limits, however, are set constructively and otherwise to the number of elements per unit length of the channel; and the most economical design can only be evolved through long experience.

Quite apart from any mechanical features of the device the character of the impulses has a decided influence on its performance and the best results will be secured, when there are produced at 4, sudden variations of pressure in relatively long intervals, while a constant pressure is maintained at 5. Such is the case in one of its most valuable industrial applications which will be specifically described.

In order to conduce to a better understanding, reference may first be made to Fig. 3 which illustrates another special use and in which 8 is a piston fixed to a shaft 9 and fitting freely in a cylinder 10. The latter is closed at both ends by flanged heads 11 and 12 having sleeves or stuffing boxes 13 and 14 for the shaft. Connection between the two compartments, 15 and 16, of the cylinder is established through a valvular conduit and each of the heads is similarly equipped. For the sake of simplicity these devices are diagrammatically shown, the solid arrows indicating the direction of undisturbed flow. An extension of the shaft 9 carries a second piston 17 accurately ground to and sliding easily in a cylinder 18 closed at the ends by plates and sleeves as usual. Both piston and cylinder are provided with inlet and outlet ports marked, respectively, 19 and 20. This arrangement is familiar, being representative of a prime mover of my invention, termed "mechanical oscillator", with which it is practicable to vibrate a system of considerable weight many thousand times per minute.

Suppose now that such rapid oscillations are imparted by this or other means to the piston 8. Bearing in mind the proceeding, the operation of the apparatus will be understood at a glance. While moving in the direction of the solid arrow, from 12 to 11, the piston 8 will compress the air or other medium in the compartment 16 and expel it from the same, the devices in the piston and head 11 acting, respectively, as closed and open valves. During the movement of the piston in the opposite direction, from 11 to 12, the medium which has meanwhile filled the chamber 15 will be transferred to compartment 16, egress being prevented by the device in head 12 and that in the piston allowing free passage. These processes will be repeated in very quick succession. If the nipples 4 and 5 are put in communication with independent reservoirs, the oscillations of the piston 8 will result in a compression of the air at 4 and rarefaction of the same at 5. Obviously, the valvular channels being turned the other way, as indicated by dotted lines in the lower part of the figure, the opposite will take place. The devices in the piston have been shown merely by way of suggestion and can be dispensed with. Each of the chambers 15 and 16 being connected to two conduits as illustrated, the vibrations of a solid piston as 8 will have the same effect and the machine will then be a double acting pump or compressor. It is likewise unessential that the medium should be admitted to the cylinder through such devices for in certain instances ports, alternately closed and opened by the piston, may serve the purpose. As a matter of course, this novel method of propelling fluids can be extended to multistage working in which case a number of pistons will be employed, preferably on the same shaft and of different diameters in conformity with well established principles of mechanical design. In this way any desired ratio of compression or degree of rarefaction may be attained.

Fig. 4 exemplifies a particularly valuable application of the invention to which reference has been made above. The drawing shows in vertical cross section a turbine which may be of any type but is in this instance one invented and described by me and supposed to be familiar to engineers. Suffice it to state that the rotor 21 of the same is composed of flat plates which are set in motion through the adhesive and viscous action of the working fluid, entering the system tangentially at the periphery and leaving it at the center. Such a machine is a thermodynamic transformer of an activity surpassing by far that of any other prime mover, it being demonstrated in practice that each single disk of the rotor is capable of performing as much work as a whole bucket-wheel. Besides, a number of other advantages, equally important, make it especially adapted for operation as an internal combustion motor. This may be done in many ways, but the simplest and most direct plan of which I am aware is the one illustrated here. Referring again to the drawing, the upper part of the turbine casing 22 has bolted to it a separate casting 23, the central cavity 24 of which forms the combustion chamber. To prevent injury through excessive heating a jacket 25 may be used, or else water injected, and when these means are objectionnable recourse may be had to air cooling, this all the more readily as very high temperatures are practicable. The top of casting 23 is closed by a plate 26 with a sparking or hot wire plug 27 and in its sides are screwed two valvular conduits communicating with the central chamber 24. One of these is, normally, open to the atmosphere while the other connects to a source of fuel supply as a gas main 28. The bottom of the combustion chamber terminates in a suitable nozzle 29 which consists of separate piece of heat resisting material. To regulate the influx of the explosion constituents and secure the proper mixture the air and gas conduits are equipped, respectively, with valves 30 and 31. The exhaust openings 32 of the rotor should be in communication with a ventilator, preferably carried on the same shaft and of any suitable construction. Its use, however, while advantageous, is not indispensable the suction produced by the turbine rotor itself being, in some cases at least, sufficient to insure proper working. This detail is omitted from the drawing as unessential to the understanding.

But a few words will be needed to make clear the mode of operation. The air valve 30 being open and sparking established across terminals 27, the gas is turned on slowly until the mixture in the chamber 24 reaches the critical state and is ignited. Both the conduits behaving, with respect to efflux, as closed valves, the products of combustion rush out through the nozzle 29 acquiring still greater velocity by expansion and, imparting their momentum to the rotor 21, start it from rest. Upon the subsidence of the explosion the pressure in the chamber sinks below the atmospheric owing to the pumping action of the rotor or ventilator and new air and gas is permitted to enter, cleaning the cavity and channels and making up a fresh mixture which is detonated as before, and so on, the successive impulses of the working fluid producing an almost continuous rotary effort. After a short lapse of time the chamber becomes heated to such a degree that the ignition device may be shut off without disturbing the established régime. This manner of starting the turbine involves the employment of an unduly large combustion chamber which is not commendable from the economic point of view, for not only does it entail increased heat losses but the explosions cannot be made to follow one another with such rapidity as would be desirable to insure the best valvular action. When the chamber is small an auxiliary means for starting, as compressed air, may be resorted to and a very quick succession of explosions can then be obtained. The frequency will be the greater the stronger the suction, and may, under certain conditions, reach hundreds and even thousands per second. It scarcely need be stated that instead of one several explosion chambers may be used for cooling purposes and also to increase the number of active pulses and the output of the machine.

Apparatus as illustrated in Fig. 4 presents the advantages of extreme simplicity, cheapness and reliability, there being no compressor, buckets or troublesome valve mechanism. It also permits, with the addition of certain well-known accessories, the use of any kind of fuel and thus meets the pressing necessity of a self-contained, powerful, light and compact internal combustion motor for general work. When the attainment of the highest efficiency is the chief object, as in machines of large size, the explosive constituents will be supplied under high pressure and provision made for maintaining a vacuum at the exhaust. Such arrangements are quite familiar and lend themselves so easily to this improvement that an enlargement on this subject is deemed unnecessary.

The foregoing description will readily suggest to experts modifications both as regards construction and application of the device and I do not wish to limit myself in these respects. The broad underlying idea of the invention is to permit the free passage of a fluid through a channel in the direction of the flow and to prevent its return through friction and mass resistance, thus enabling the performance of valve functions without any moving parts and thereby extending the scope and usefulness of an immense variety of mechanical appliances.

I do not claim the methods of and apparatus for the propulsion of fluids and thermodynamic transformation of energy herein disclosed, as these will be made subjects of separate applications.

I am aware that asymmetrical conduits have been constructed and their use proposed in connection with engines, but these have no similarity either in their construction or manner of employment with my valvular conduit. They were incapable of acting as valves proper, for the fluid was merely arrested in pockets and deflected through 90°, this result having at best only 25% of the efficiency attained in the construction herein described. In the conduit I have designed the fluid, as stated above, is deflected in each cycle through 360°, and a co-efficient approximating 200 can be obtained so that the device acts as a slightly leaking valve, and for that reason the term "valvular" has been given to it in contrast to asymmetrical conduits, as heretofore proposed, which were not valvular in action, but merely asymmetrical as to resistance.

Furthermore, the conduits heretofore constructed were intended to be used in connection with slowly reciprocating machines, in which case enormous conduit-length would be necessary, all this rendering them devoid of practical value. By the use of an effective valvular conduit, as herein described, and the employment of pulses of very high frequency, I am able to condense my apparatus and secure such perfect action as to dispense successfully with valves in numerous forms of reciprocating and rotary engines.

The high efficiency of the device, irrespective of the character of the pulses, is due to two causes: first, rapid reversal of direction of flow and, second, great relative velocity of the colliding fluid columns. As will be readily seen each bucket causes a deviation through an angle of 180°, and another change of 180° occurs in each of the spaces between two adjacent buckets. That is to say, from the time the fluid enters or leaves one of the recesses to its passage into, or exit from, the one following a complete cycle, or deflection through 360°, is effected. Observe now that the velocity is but slightly reduced in the reversal so that the incoming and deflected fluid columns meet with a relative speed, twice that of the flow, and the energy of their impact is four times greater than with a deflection of only 90°, as might be obtained with pockets such as have been employed in asymmetrical conduits for various purposes. The fact is, however, that in these such deflection is not secured, the pockets remaining filled with comparatively quiescent fluid and the latter following a winding path of least resistance between the obstacles interposed. In such conduits the action cannot be characterized as "valvular" because some of the fluid can pass almost unimpeded in a direction opposite to the normal flow. In my construction, as above indicated, the resistance in the reverse may be 200 times that in the normal direction. Owing to this a comparatively very small number of buckets or elements is required for checking the fluid. To give a concrete idea, suppose that the leak from the first element is represented by the fraction $\frac{1}{X}$, then after the nth bucket is traversed, only a quantity $\left(\frac{1}{X}\right)^n$ will escape and it is evident that X need not be a large number to secure a nearly perfect valvular action.

What I claim is:

1. A valvular conduit having interior walls of such conformation as to permit the free passage of fluid through it in the direction of flow but to subject it to rapid reversals of direction when impelled in the opposite sense and thereby to prevent its return by friction and mass resistance.

2. A valvular conduit composed of a closed passageway having recesses in its walls so formed as to permit a fluid to pass freely through it in the direction of flow, but to subject it to rapid reversals of direction when impelled in an opposite sense and thereby interpose friction and mass resistance to the return passage of the same.

3. A valvular conduit composed of a tube

or passageway with rigid interior walls formed with a series of recesses or pockets with surfaces that reverse a fluid tending to flow in one direction therein and thereby check or prevent flow of the fluid in that direction.

4. A valvular conduit with rigid interior walls of such character as to offer substantially no obstacle to the passage through it of fluid impulses in one direction, but to subject the fluid to rapid reversals of direction and thereby oppose and check impulses in the opposite sense.

5. A valvular conduit with rigid interior walls formed to permit fluid impulses under pressure to pass freely through it in one direction, but to subject them to rapid reversals of direction through 360° and thereby check their progress when impelled in the opposite sense.

A valvular conduit with rigid interior walls which permit fluid impulses to flow through it freely in one direction, formed at a plurality of points to reverse such fluid impulses when impelled in the opposite direction and check their flow.

7. A valvular conduit with rigid interior walls having pockets or recesses, and transversely inclined intermediate baffles to permit the free passage of fluid impulses in one direction but to deflect and check them when impelled in the opposite direction.

In testimony whereof I affix my signature
NIKOLA TESLA.

N. TESLA.
VALVULAR CONDUIT.
APPLICATION FILED FEB. 21, 1916. RENEWED JULY 8, 1919.

1,329,559. Patented Feb. 3, 1920.

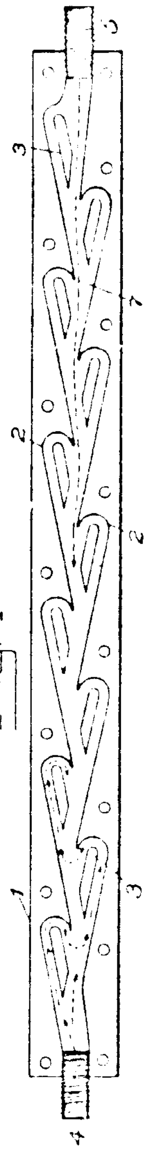

Fig. 1

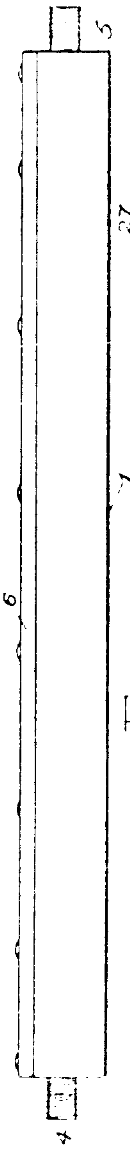

Fig. 2

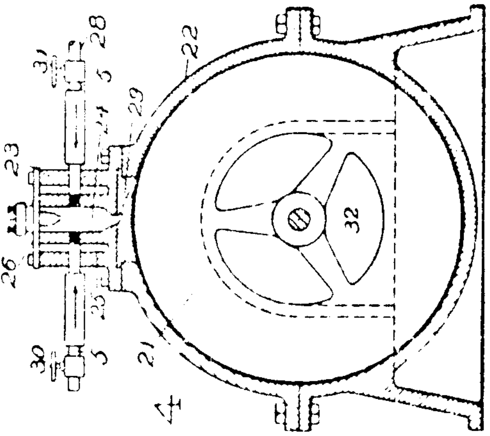

Fig. 4

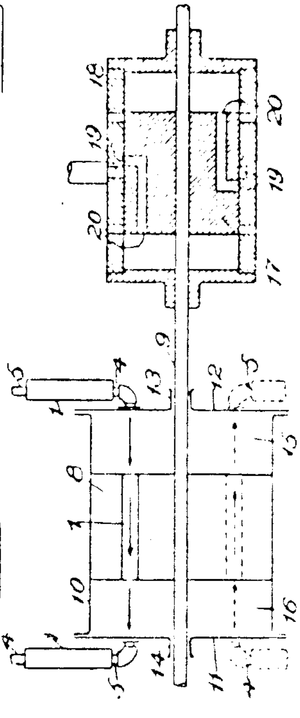

Fig. 3

INVENTOR
Nikola Tesla
BY
ATTORNEY

A Guide to Fluidics, ed. Arthur Conway
McDonald & co, London 1971, p. 76, 77, 135
"Nuclear Applications" by M. J. Fisher

5.2 Diodes

Three types of diode have received attention for reactor applications.

Probably the best known and simplest of them is the *vortex diode*, described in Sections 1.8 and 2.4.2. The device is in its high-resistance state when fluid flows in through the tangential port, forms a vortex in the chamber and leaves through the axial port. With flow in the other direction the device is in its low-resistance state—a vortex is not formed.

The performance of diodes is usually quoted as the ratio of the pressure drops for the same flow in the two directions, though sometimes the flow ratio for the same pressure drop is quoted. The former is approximately equal to the square of the latter.

Although fluidics is thought of as a new technology the earliest publication on diodes was made in 1920 (*5*). The first report on vortex diodes, made in 1935 (*6*), quoted pressure ratios of up to about 40 : 1. More recent work quotes ratios of 39 : 1 (*1*) and 50 : 1 (*7*). Higher ratios could be achieved but only by using excessively large vortex chambers.

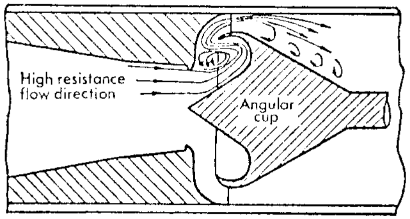

Figure 5.3 Scroll diode (*7*).

The input and output lines are at right angles, which could be an advantage in some applications. In general parallel input and output lines are preferred, so a right-angled bend must be made in the pipework leading to the diode.

Cavitation affects performance. In the forward direction cavitation occurs in the tangential port, which increases resistance and reduces the effective pressure ratio. This can be important where the working pressure is close to the vapour pressure.

The *cup* or *scroll diode* (fig. 5.3) has a geometry more suitable for in-line pipe arrangement. Here a larger annular vortex is formed in the reverse direction. The performance is poor. Pressure ratios reported are about 8 : 1 (*1*) and 6·6 : 1 (*7*).

Another in-line device is known as the *fluid flow rectifier* or *cascade diode* (fig. 5.4). (The terms 'rectifier' and 'diode' illustrate that the antecedents of those who develop fluidics are as much in electrical as in mechanical engineering.) The device has many stationary blades, shaped like turbine blades, attached to a rod fixed along the axis of a tube. The blades are positioned so that if fluid enters from the left it is deflected and is caused to spin. After it has passed the first few blades the flow velocity is almost at right angles to the tube axis. The vortex induces a large pressure loss, and the device is in its high-resistance state. Flow in the other direction meets the device in its low-resistance state, for the blades present a bluff obstacle. The fluid does not spin but passes almost axially down the tube between the rows of blades.

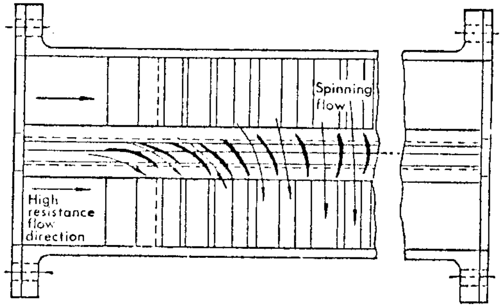

Figure 5.4 Cascade diode (*7*).

Pressure ratios as high as 181 : 1 have been reached (*8*) and higher ratios can be achieved.

Fluid flow rectifiers are less prone to cavitation than vortex diodes. In fact their performance is better than that of any other fluidic diode. But they are relatively complex, and their expense can only be justified in critical areas.

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Chapter 5

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Jan. 5, 1932. D. THOMA 1,839,616

FLUID LINES

Filed May 24, 1929

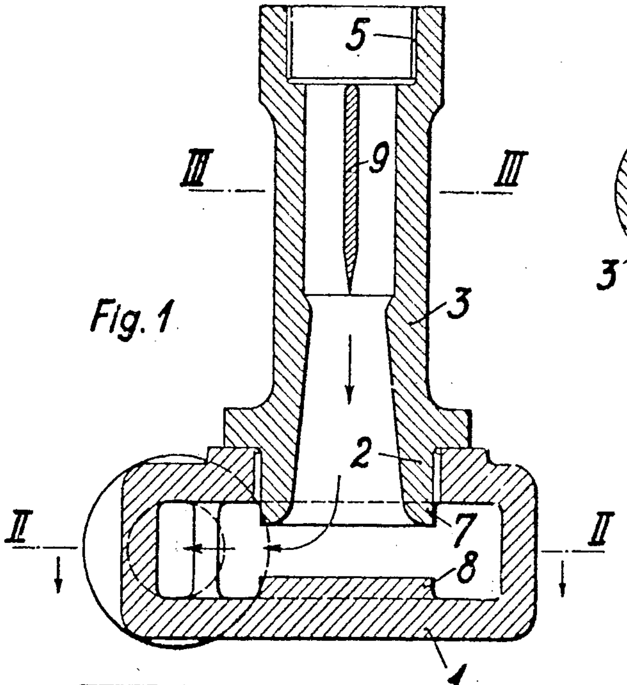

Fig. 1

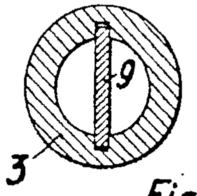

Fig. 3

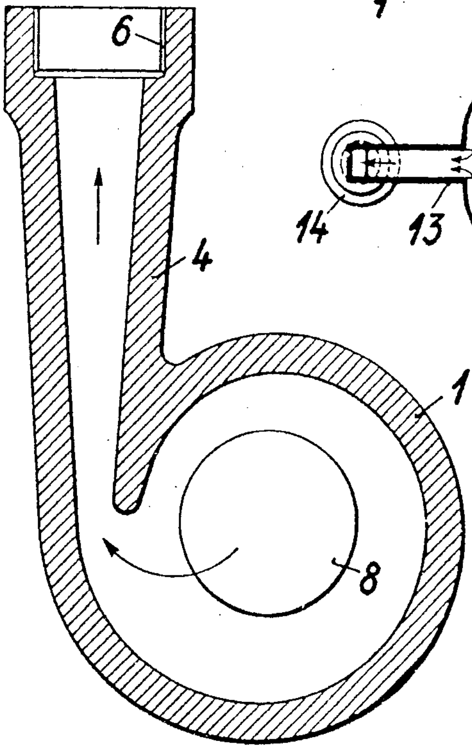

Fig. 2

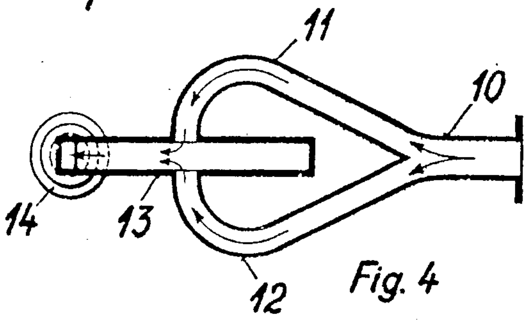

Fig. 4

Inventor
Dieter Thoma
By Cumpston + Griffith
his Attorneys

Patented Jan. 5, 1932 1,839,616

UNITED STATES PATENT OFFICE

DIETER THOMA, OF MUNICH, GERMANY

FLUID LINES

Application filed May 24, 1929, Serial No. 365,753, and in Germany June 11, 1928.

The invention relates to fluid lines for conducting gaseous or liquid fluid and has for its object to establish a practically unidirectional flow in such a line in a very simple and efficient manner without using any movable parts.

Check valves built into a line permit a flow in only one direction, but these mechanisms share the disadvantage incidental to moving parts—liability of becoming clogged, damaged or partly destroyed by the fluid passing through—and are therefore apt to come out of order.

There is also the danger of a hammering action, the sudden closure causing frequently an undue increase of pressure in the line. In some cases especially when conducting a fluid of light weight under a small head—gases—it proves quite a difficult problem to provide a check valve working satisfactorily at all.

According to my invention I impose a great resistance to a flow in one direction while the flow in the opposite direction is practically not restricted. For this purpose I provide a direct path for the fluid flowing in one direction and conduct the fluid through a circular or partly circular course when flowing in the opposite direction. I furthermore arrange the intake closer to the center of this circular path than the outlet and locate the latter preferably tangential to the circular course. The ratio of resistance in one direction to the resistance in the opposite direction may be varied in wide limits according to requirements.

The above and various other improvements will be hereinafter fully described and claimed. In order that this my invention may be more readily understood reference is made to the accompanying sheet of drawings which show how the invention may be carried into practical effect.

Fig. 1 shows a device according to my invention in section.

Fig. 2 is a section taken on line II—II of Figure 1.

Fig. 3 is a section taken on line III—III of Figure 1.

Fig. 4 is a modification in diagrammatic form of the device shown in the Figures 1 and 2.

The normal direction of flow is indicated by arrows on the drawings. A housing 1 of approximately circular or spiral cross section is provided with a threaded opening 2 into which the inlet 3 is screwed. A tangential outlet 4 is part of the housing. The outlet 4 is preferably made with a gradually increasing cross section. The inlet 3 is also provided with a cross section gradually increasing in the direction of the flow. The ends of the inlet and outlet are threaded at 5 and 6 for building the device conveniently into a fluid line.

With a device dimensioned aproximately in scale to the Figures 1 and 2 the resistance against fluid passing through in the direction of the arrows is less than the resistance offered by an ordinary check valve while fluid passing through in the opposite direction finds a vastly (in the present case about 35 times) greater resistance. This is due to the circular whirl being formed by the fluid when passing through in a direction opposite to the arrows the resulting centrifugal forces causing a considerable difference of pressure in the two connections 3 and 4. In order to facilitate the forming of a whirl a ring shaped projection 7 of the inlet 3 and a circular shoulder 8 of the housing 1 protrude from opposite sides into the chamber giving the latter the shape of an annular ring with the outlet branching tangentially therefrom. The velocity of the fluid in the whirl is influenced by the tangenital connection 4 being formed as a nozzle.

In order to prevent a turbulent flow of the fluid when streaming in a direction opposite to the arrows on the drawings there is provided a stabilizing plate 9 placed in grooves of the central connection 3 (see Fig. 3).

The arrangement shown diagrammatically in Figure 4 provides an inlet 10 branching into two lines 11 and 12 entering axially from opposite sides into the housing 13. This housing 13 and the outlet 14 may be the same as the housing 1 and the outlet

4 described before. This arrangement permits the use of a smaller size of inlet whereby the resistance against a flow in a direction opposite to the arrows may be greatly increased.

It is furthermore immaterial whether a single or a plurality of outlets are employed. Having now fully explained my invention I do not wish to be understood as limiting myself to the exact details of construction nor to the particular applications shown, as obviously many modifications will occur to a person skilled in the art.

What I claim is:

1. A device for resisting reverse flow in fluid lines, comprising a housing containing a chamber of approximately circular shape, said chamber being free from vanes and substantially unobstructed, an inlet close to the center of said chamber, an outlet approximately tangential to said chamber, so that fluid flowing in through said inlet may flow directly and with low resistance to and through said outlet and so that fluid flowing in through said outlet in a reverse direction will rotate in said chamber and encounter substantial resistance to such reverse flow, and guiding means built into said inlet for preventing a turbulent movement of the fluid in case of a flow opposite to the normal direction.

2. A device for resisting reverse flow in fluid lines, comprising walls forming a chamber having a thicker annular portion and a thinner central portion, said chamber being free from vanes and substantially unobstructed, an inlet connection leading to said central portion, and an outlet connection leading in a generally tangential direction from said annular portion, so that fluid entering said chamber from said inlet connection may flow substantially directly and with relatively low resistance to said outlet connection and so that fluid entering from said outlet connection will be caused to whirl in said chamber and may flow to said inlet connection only with relatively high resistance.

3. The method of retarding reverse flow which comprises interposing a chamber in a fluid line, causing fluid flowing in one direction in said line to flow substantially directly and unimpededly through said chamber, and causing fluid flowing in the opposite direction to whirl in said chamber with substantial rotary movement to impede reverse flow to a substantial extent.

4. The method of retarding reverse flow which comprises interposing in a fluid line a rounded chamber, introducing fluid flowing in one direction in the fluid line into the chamber substantially centrally thereof, and causing the introduced fluid to flow substantially directly and unimpededly through said chamber to substantially tangential outlet and thence to the fluid line, so that if fluid attempts to flow in said line in a reverse direction it will enter said chamber by said tangential outlet and will be caused thereby to rotate in said chamber to produce substantial impediment to such reverse flow.

In testimony whereof I hereunto affix my signature.

DIETER THOMA.

TRANSACTIONS OF THE MUNICH HYDRAULIC INSTITUTE

Bulletin No. 3
p. 13-28

An Investigation of the Thoma Counterflow Brake

By RICHARD HEIM[1]

Translated by Morrough P. O'Brien[2]

CHECK valves of the usual type are unsatisfactory for many conditions, since they may fail completely at the instant of operation because of sticking, or close too quickly, causing water hammer. A device having no moving parts and showing very different resistances in the two directions of flow would offer great advantages, although naturally it could not fully prevent the back flow. Prof. D. Thoma has suggested an apparatus, "the counterflow brake," consisting of a spiral vortex chamber with tangential and axial connections (Fig. 1). Because of the tangential approach of the water and the resulting vortex, the apparatus has a great resistance in this direction of flow (solid arrows in Fig. 1), while the flow in the opposite direction (broken arrows in Fig. 1) encounters a far smaller resistance.

Since the characteristics of the flow in the direction of greatest resistance are most interesting in these investigations, we will, in the following discussion, regardless of the fact that flow occurs in the opposite direction in this valve, refer to the tangential connection as the supply line and the axial connection as the discharge line.

SCOPE OF THE WORK

The particular scope of this work was to investigate the manner of flow and the pressure distribution with the suggested counterflow brake, employing the usual hydraulic methods of measurement, and to experiment on its operation with the alteration in design suggested by Prof. D. Thoma, so as to work out the most serviceable form of the valve for the purpose and to develop several practical types and define their range of application.

CONSIDERATION OF THE MANNER OF FLOW IN THE SPIRAL VORTEX CHAMBER

So as to obtain a visual picture of the manner of flow within the spiral casing, the apparatus shown in Fig. 2 was cut from dry "pattern wood" and fastened together by screws and glue; the joints were filled with putty and painted over with lacquer. The cover over the glass plate could be removed so that it would not be cracked by warping between experimental periods. A closed, wooden supply pipe, 3 m. in length, was inserted into the spiral casing and then made watertight, thus insuring a smooth transition from the straight channel. At *a*, a bronze nozzle on which the discharge pipe could be screwed was inserted into the wood.

An interesting phenomenon appeared during flow through the channel; at the center of the vortex, a space devoid of water formed directly over the discharge opening and extended close up to the glass plate and down into the discharge pipe. This cavity had approximately the shape sketched in Fig. 3. A photographic record of this phenomenon was not possible because of the difficult lighting conditions.

An attempt was made to prevent the occurrence of this cavity by arranging the apparatus with the discharge pipe directed vertically upward, but this change was without effect. A temporary closure of the discharge pipe did not alter the conditions in the spiral.

The volume devoid of water must be regarded as a vacuum, or better, as a volume sparsely filled with air derived from the water and water vapor; the decrease in pressure within the spiral is so great that the pressure, at the center of the spiral, falls even as low as the vapor pressure, if the entrance velocity of the water is sufficiently high, resulting in "cavitation."

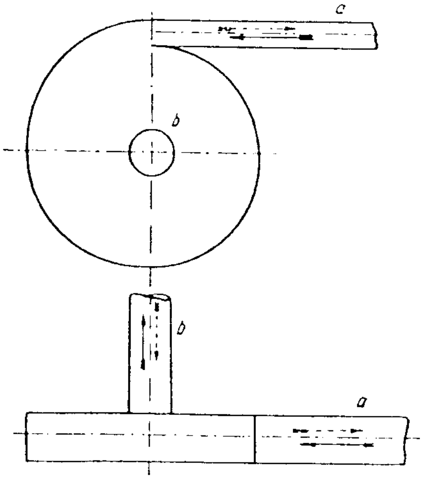

Fig. 1

Attempts at determining the streamlines with woolen strings failed because the streamers became tangled and hung down into the discharge pipe without indicating the paths. The introduction of sawdust was difficult and unsuccessful as was true of potassium permanganate; the sawdust and coloring, which diffused too quickly, appeared as clouds in the bottom opening but did not clearly show the individual streamlines.

A mixture of tetralin and chloroform highly colored with indigo was found to be very satisfactory for indicating the streamlines in closed channels of high velocities, since this oily mixture does not dissolve in water. The ingredients were compounded in such a manner that the mixture had a specific gravity equal to that of water. The mixture was inserted through a nozzle at *b* (Fig. 2) and formed small drops about 1 to 2 mm. in diameter, which clearly showed the flow over the entire height of the chamber. The incoming water (Fig. 4) follows the guiding wall of the spiral and flows in toward the discharge opening with ever increasing angular velocity. It was observed that drops of the colored mixture which were injected approached the discharge opening at a steady rate and then remained rotating in the vicinity of the opening for some time without altering their distance from the center of the vortex. They then sprang outward to about halfway between the center and wall of the spiral and finally, after a very long time, were carried downward. In Fig. 4, the path of a small drop is shown by a dotted line; the shaded area represents the cavity at the center of the spiral.

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[1] Köln, Germany. Formerly graduate student assistant at the Munich Hydraulic Institute. This paper is the thesis submitted by Dr. Heim for his doctor's degree.

[2] Associate Professor of Mechanical Engineering, University of California, Berkeley, Calif. John R. Freeman Scholar, A.S.C.E., 1927-1928. Assoc-Mem. A.S.M.E.

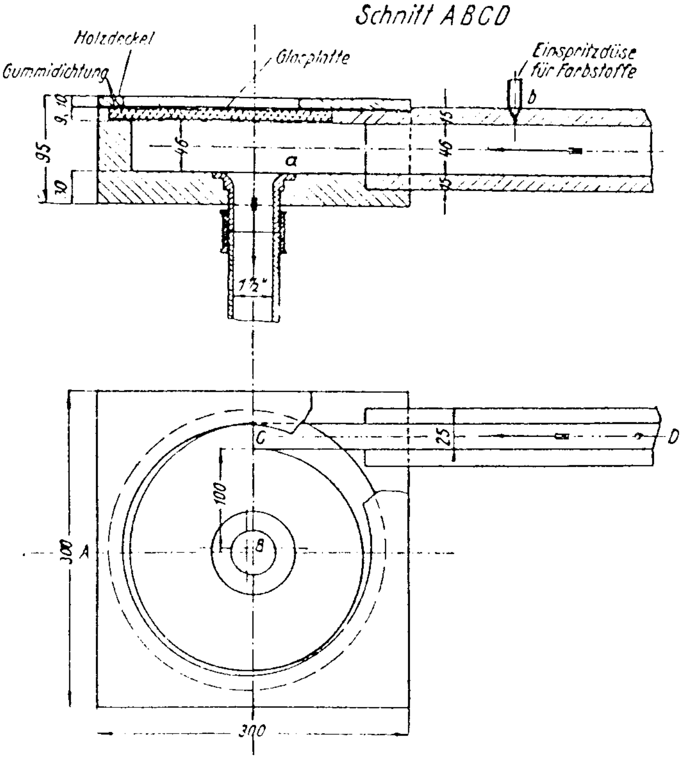

FIG. 2

(*Holzdeckel* = wooden cover plate; *Gummidichtung* = rubber seal; *Schnitt* = section; *Glasplatte* = glass plate; *Einspritzdüse für Farbstoffe* = injection nozzle for coloring matter.)

The sudden backward motion of the individual rotating particles in the center of the vortex permits the conclusion that, within the spiral in a plane perpendicular to the plane of the principal rotation, there is a kind of secondary eddy, such as is shown by the arrows in Fig. 3. The layers of water adjacent to the upper and lower sides experience a braking effect, due to the skin friction, and the resulting reduction in the centrifugal force causes a flow toward the center of the vortex. The fluid particles then flow around the vortex until caught by the faster moving layers farther from the walls and are moved away from the discharge opening by the rapidly increasing centrifugal forces. In this manner, there arises centrifugal eddy flow in the vicinity of the top and bottom and centripetal flow at the mid-point between them.

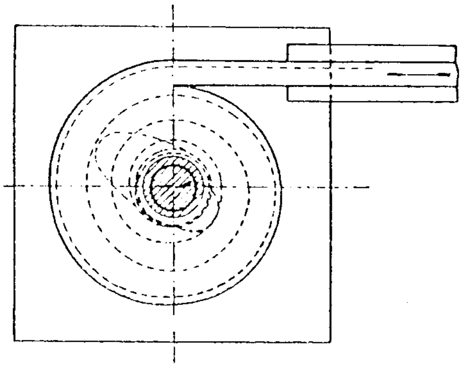

FIG. 4

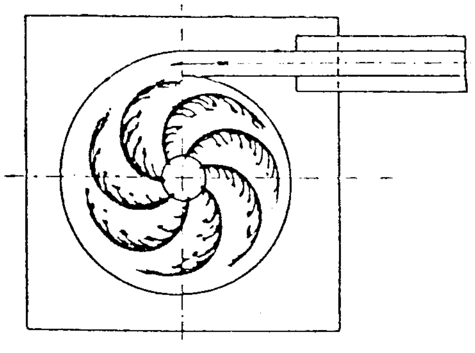

FIG. 5

That the layers near the top and bottom, which are impeded in their rotational velocity, are strongly driven toward the discharge pipe can be clearly observed at low velocities of the approaching stream. By means of crystals of potassium permanganate strewn over the bottom of the spiral, the currents in the bottom layer can be separated from the layers in the interior of the chamber, for velocities which do not remove the crystals instantly. It was shown by this method that the bottom layers flow toward the discharge opening in star-shaped paths as appears in Fig. 5, while the upper layers exhibit the same streamlines as are shown in Fig. 4. However, no cavity was formed in this experiment.

When the wooden apparatus was constructed, no thought was given to its applicability for higher pressures and the wooden construction quickly became leaky because of the pressure and warping and was unfit for quantitative measurements. An apparatus similar to this one in principle was constructed of brass plates and sheet metal. An exact description will be given later. In order to clarify the nature of the cavity at the center of the spiral casing, this brass spiral was arranged as is shown in Fig. 6. A glass tube about 800 mm. long was placed between the spiral and a tank 300 mm. high and 500 mm. in diameter. At a supply pressure in the tangential pipe of 17 m. of water, determined by the head-control tank, the entire content of the upper tank usually started to rotate during the initiation of flow, so that a free surface vortex formed, previous to the use of a float, through which air could be sucked into the spiral. By using a float, the intake of air was prevented and the phenomenon sketched in Fig. 6 was evident. Cavities of this form were evident in the glass tube in steady upward and downward motion, alternately changing from being drawn downward by the vacuum to being moved upward by buoyancy.

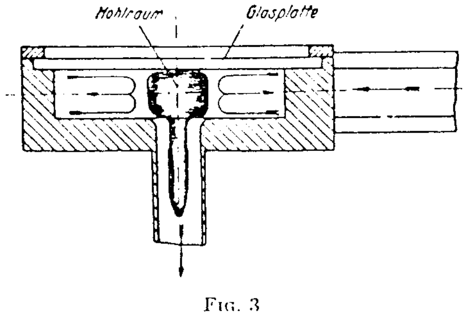

FIG. 3

(*Hohlraum* = cavity; *Glasplatte* = glass plate.)

At *a* (Fig. 6), a hole was drilled for the measurement of static pressure. If this hole was opened, so much air was drawn through it that the glass tube quickly appeared to be filled with air and only a very thin surface layer of water spiraled up to the tank. After the hole was closed, this air column dropped back and formed an air pocket which occupied half of the clear distance between the spiral casing and the tank. The cavity retained this strength. Wooden sticks of any size introduced into the glass tube from above were drawn downward and held by the vacuum. During all of these measurements, the discharge, as given approximately by stop-watch and scales, did not change appreciably with varying supply pressure.

Fig. 7 shows a photograph of the development of cavitation from its initial condition to full strength in the form of a twisted column. If the gate valve before the spiral was opened suddenly and the velocity of the flow increased as rapidly as possible, carefully preventing the introduction of air at *a* as well as from the chamber, the water column in the glass tube remained clear for a short time and then suddenly became milky with a cloud of small bubbles of air liberated from the water. These bubbles formed simultaneously immediately above the axial outlet of the spiral and below the tank (Fig. 6). After a fraction of a second, the fine mist in the tube collected into a streak in the center which connected the distinct fog above the spiral and below the tank. After another two seconds, the general appearance was as is shown in Fig. 7, Photograph 1. The spindle-shaped vacuum was drawn slowly upward with an occasional backward motion, and it was evident that in the vicinity of the axis only very slight axial velocities occurred. After 30 sec., the cavity had taken on the form shown in Photograph 2. The last picture (3) illustrates the steady condition attained one minute after the valve had been fully opened.

DESCRIPTION OF THE EXPERIMENTAL EQUIPMENT

The greater durability and tightness of the new metal spiral permitted higher velocities and the use of pressures up to 6 atmospheres. This apparatus was so arranged as to permit changes in construction without great difficulty. Fig. 8 shows the complete apparatus to scale as it was used in the experiments to be described. It was constructed of $1^1/_2$-in. gas pipe and fittings. By various arrangements of the valves 1 to 4, the water entering at *a* and discharging into the weighing tank at *b* could be directed through the spiral vortex chamber in either of the two directions of flow.

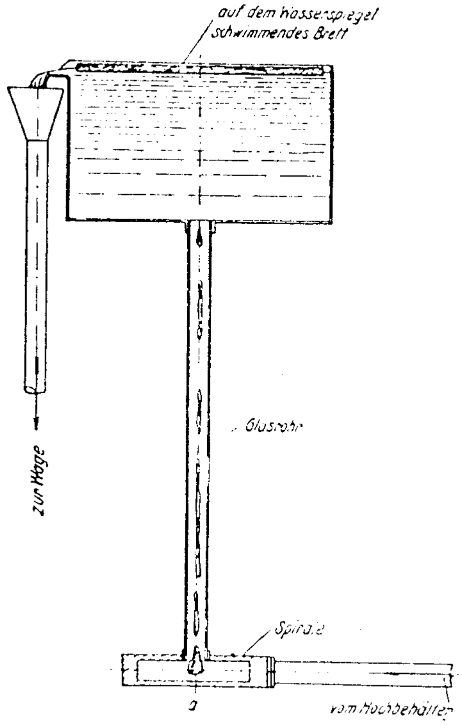

FIG. 6

(*auf dem Wasserspiegel schwimmendes Brett* — plank floating on water surface; *zur Wage* = to scales; *Glasrohr* = glass tube; *Spirale* = spiral vortex chamber; *vom Hochbehalter* = from head-control tank.)

The experimental apparatus was placed in the basement of the laboratory and was under a head of 17 m. provided by the

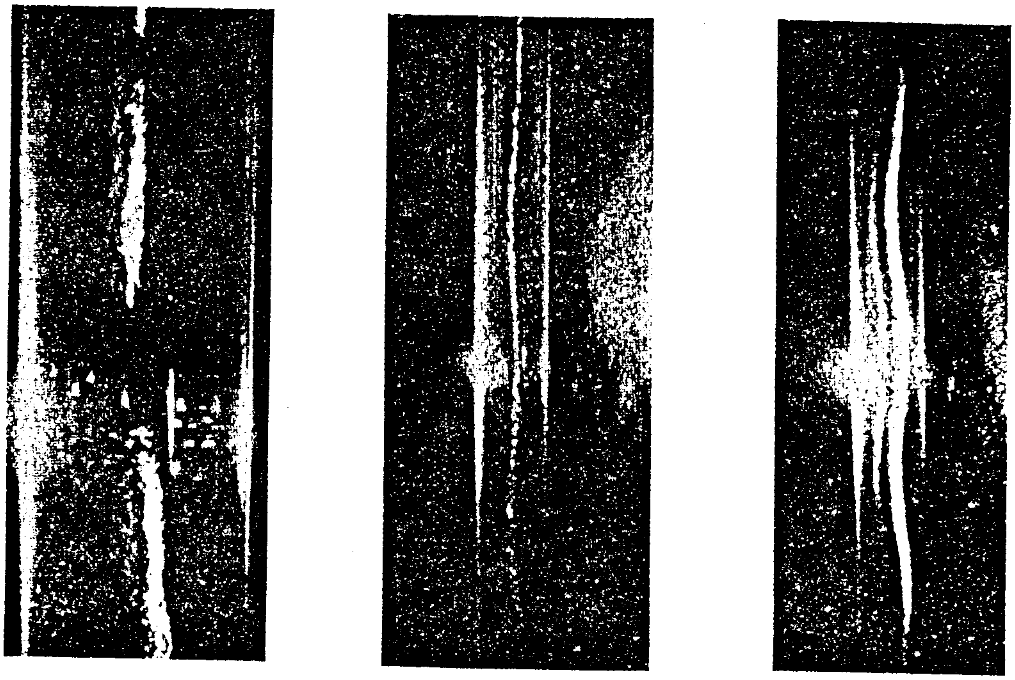
1 2 3

FIG. 7

head-control tank. The water surface in the head control was held almost constant by a proper supply and an overfall. A connection to the city water supply gave a static pressure of 4 or 5 atmospheres, but considerable and frequent variations of pressure occurred in this line, and the measurements carried out with the city pressure served only as an approximate measurement in a certain series of experiments described later.

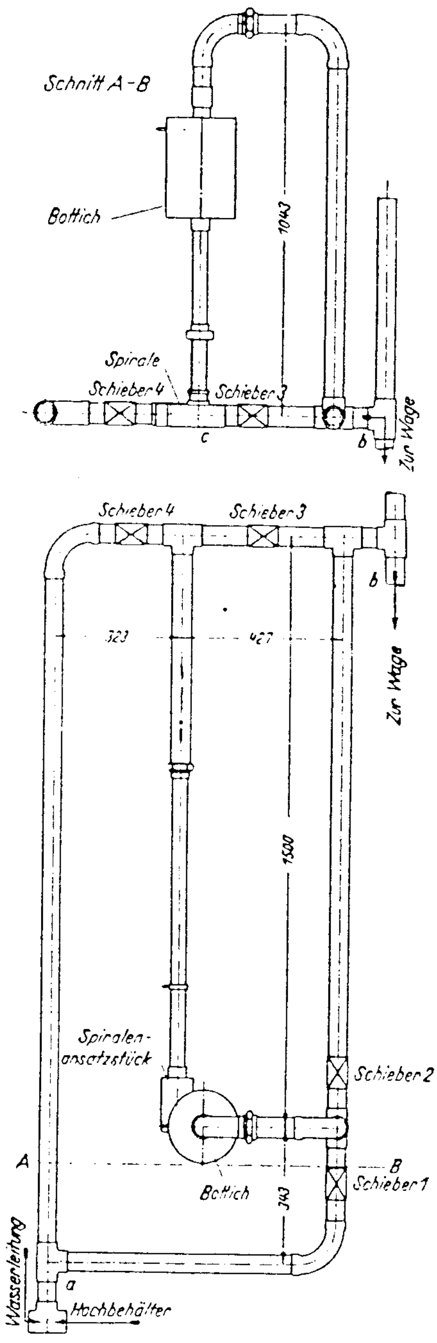

FIG. 8

[*Schnitt* = section; *Bottich* = tank (Stilling Chamber No. 1); *Spirale* = spiral vortex chamber; *Schieber* = valve; *zur Wage* = to scales; *Spiralenansatzstück* = tangential inlet to spiral vortex chamber; *Wasserleitung* = city water line; *Hochbehälter* = head-control tank.]

Above the spiral casing, a stilling chamber was inserted (Fig. 9), containing 2 sheet-metal plates with 6-mm. perforations separated by 21 pipes, 150 mm. long and having an internal diameter of one inch (Stilling Chamber No. 1). In place of this stilling chamber, a group of pipes, as shown in Fig. 10, was used later (Stilling Chamber No. 2).

These stilling chambers were used to guide the water, which leaves the spiral casing with a whirling component of flow, into parallel paths and to insure a correct measurement of the static pressure. The method of setting up the spiral casing and the stilling chambers appears in Figs. 11 and 12.

DESCRIPTION OF THE SPIRAL VORTEX CHAMBER

Two vortex chambers of different sizes, having outlines following a logarithmic spiral, were constructed of brass. The side walls of the larger spiral (Fig. 13) consisted of plate brass, 1 mm. thick, bent to follow the logarithmic spiral and held by a number of small metal screws. The cracks between bottom, top, and side walls were made watertight by means of red lead and gypsum. Different throat pieces cut from hard rubber could be inserted into the rectangular brass supply pipe at *b*. These pieces were polished and the head was recessed to project over the end of the supply pipe. By inserting brass shims, it was possible to set these throat pieces exactly and easily and to check them at the end of the experiment. The design was such that sudden expansions and contractions of the flow were eliminated. The axial discharge pipe was threaded so that it could be screwed down into the spiral casing.

The smaller vortex chamber (Fig. 14) was made from a single piece of brass, and the bottom and spiral wall were fitted with a flaring nozzle which held the reducer and nozzle body. A brass plate 10 mm. thick was used as a cover plate and screwed in place; in this plate, also, a pipe with a fine thread was inserted at right angles so that it could be moved to any desired depth in the cavity. The two nozzles used with this spiral formed a gradual transition from the 38-mm. circular supply pipe to a rectangular section, 24 by 38 mm. or 20 by 38 mm. From this section, the flow was further constricted by a series of interchangeable reducers (Fig. 14) to an entrance area 20 by 38 mm. or 10 by 38 mm. Also, in the case of this spiral, an impact-free transition from the tangential supply line to the vortex chamber was obtained.

CONDUCT OF THE EXPERIMENTS

Time and water measurements in the experiments were made with a stop-watch with double hands and with scales. The starting error of the watch was eliminated by using the two hands, and the error due to the sluggishness of the scales was eliminated by proper procedure in weighing. The time ranged from 30 to 60 sec., and the measuring basin held 150 liters. The necessary pressure measurements were made by means of a mercury differential manometer. For the experiments using the city water mains, capillary tubes 0.25 mm. in diameter were used to damp out the violent oscillations of the mercury. The point of connection of the manometers will be stated in the discussion of the individual experiments.

SOURCES OF ERROR

The error in the measurement of the time depended upon the observer and amounted to about 0.5 sec., which introduced an error in the rate of discharge of from 0.5 to 1.0 per cent. The sensitivity of the scales gave an error in the weight of less than 0.05 per cent.

The height of the meniscus of the mercury was estimated to the nearest 0.5 mm.; this procedure gave a possible error of 1 mm. for the two legs and resulted in an observational error of 1 per cent for a differential head of 10 cm. The preponderance

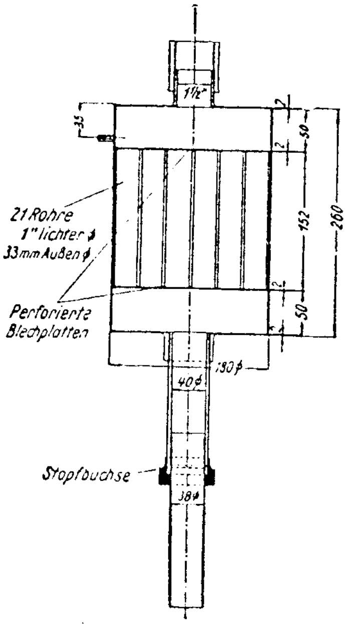

FIG. 9

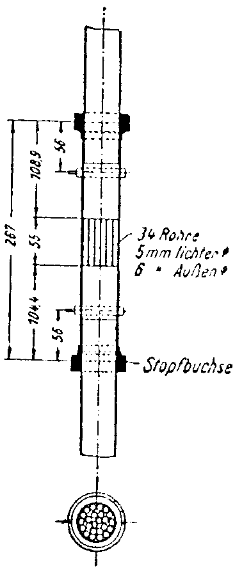

FIG. 10

FIG. 9 STILLING CHAMBER NO. 1
(*Rohre* = pipes; *lichter* = inside diameter; *Aussen* = outside diameter; *perforierte Blechplatten* = perforated sheet-metal plate; *Stopfbuchse* = stuffing-box.)

FIG. 10 STILLING CHAMBER NO. 2
(*Rohre* = pipes; *lichter* = inside diameter; *Aussen* = outside diameter; *Stopfbuchse* = stuffing-box.)

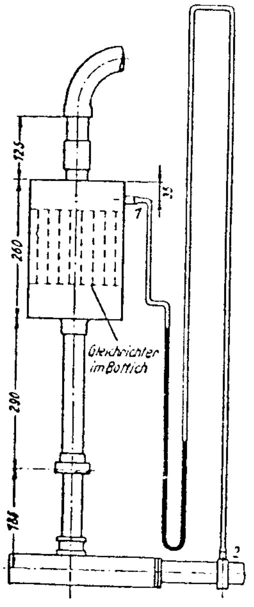

FIG. 11

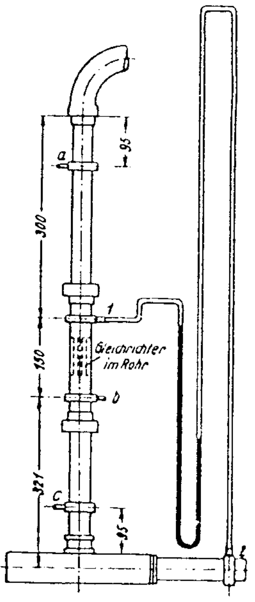

FIG. 12

FIG. 11
(*Gleichrichter im Bottich* = Stilling Chamber No. 1.)

FIG. 12
(*Gleichrichter im Rohr* = Stilling Chamber No. 2.)

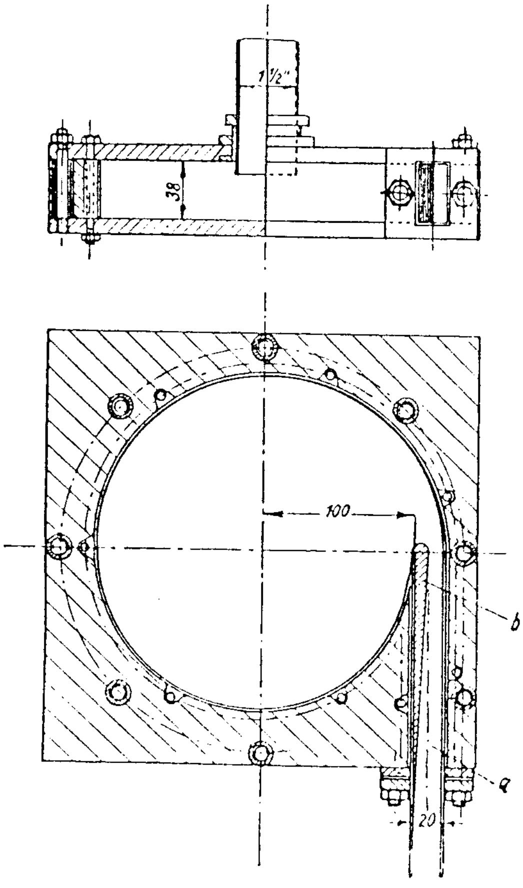

FIG. 13

of pressure differences was greater than 10 cm. and only a few were smaller.

The meniscus was in general steady, but periodic oscillations were observed for deflections between 100 and 500 cm. These oscillations had a maximum amplitude of from 3 to 5 mm. The maximum error in this case was about 1 per cent of the length of the column of mercury; pressure variations during the use of the city water supply, which were very considerable, are not considered here.

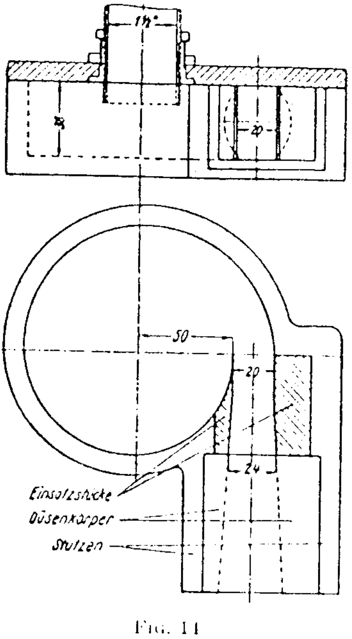

FIG. 14
(*Einsatzstücke* = reducer; *Düsenkörper* = nozzle; *Stutzen* = nozzle support.)

The only errors of any influence are those in the pressure and time measurements. The maximum values given are not in general additive, since, for low water velocities and small pressure differences, the error in the pressure measurement is of influence but that in the time is scarcely important because of the long time of weighing; the converse is true for higher velocities and greater pressure differences.

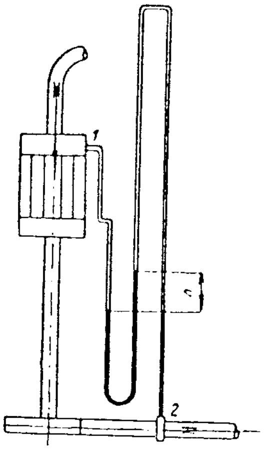

Fig. 15

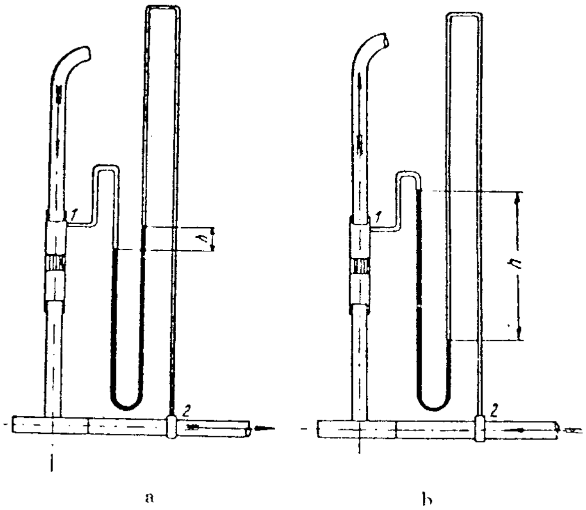

Fig. 16

The conditions became unsatisfactory with the development of pressure variations, which were limited, however, to a small number of experiments.

The accuracy attained was sufficient for the purpose of the experiments.

COMPUTATION AND PRESENTATION OF EXPERIMENTAL RESULTS

The differential head measured by the manometer had to be corrected by the velocity head (which is additive or subtractive, according to the direction of flow) in order to determine the loss of head in the spiral vortex chambers. The pressure was always measured where it was permissible to assume that eddy motion had been eliminated; the velocity head at these points was based upon the mean velocity, $v = Q/F$.

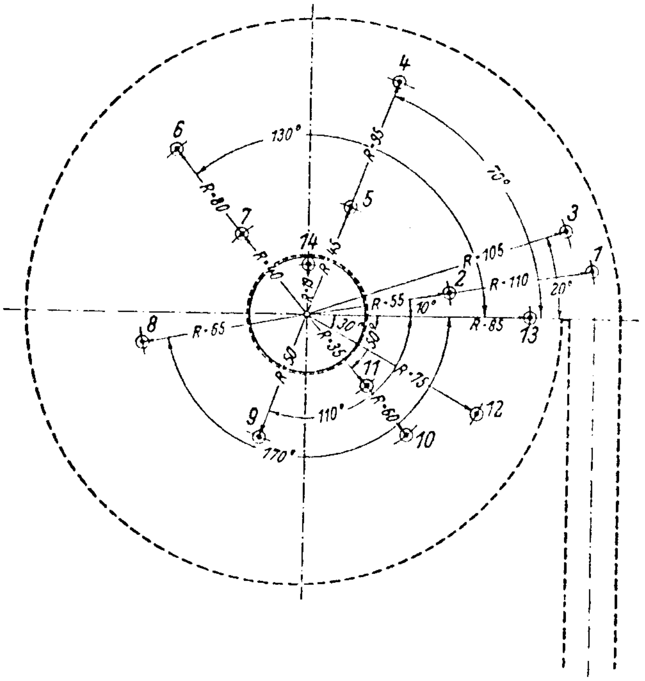

Fig. 17

The segregation of the loss in the spiral vortex chamber into its various component losses, such as those in the baffles and in the pipe lines, appeared undesirable, since an estimate of the friction loss under the different conditions of flow was difficult to make. The losses which are given in subsequent paragraphs include the friction loss in the relatively short pipe between the points of measurement and in the stilling chamber. In only a few exceptional instances is the loss in the guide vanes (*Wirbelzerstörer*) included in the resistance coefficient in place of that in the stilling chamber.

In conducting the experiments, guide vanes or similar apparatus were always used to eliminate the eddying motion and to make sure of reliable pressure measurements. Considering also the practical application of the apparatus, it was necessary to develop a design which would not discharge water, having a strong rotational component, since the law governing the resistance for this type of flow is not known. A design giving a whirling discharge would therefore be open to the criticism of being uncertain in its operation.

The arrangement of the two stilling devices is shown schematically in Figs. 15a, 15b, 16a, and 16b. A more complete description has been given previously (see page 16 and Figs. 9 and 10 on page 17).

In the following discussion, v_1 and v_2 denote the velocities at the corresponding points of measurement (Figs. 15a to 16b), while v_0 indicates the corresponding velocity in a 38-mm. pipe. For the direction of flow in Figs. 15a and 16a, the loss is given by

$$H_w = h \times 12.6 - \frac{v_2^2}{2g} + \frac{v_1^2}{2g}$$

and for Figs. 15b and 16b the loss is

$$H_w = h \times 12.6 + \frac{v_2^2}{2g} - \frac{v_1^2}{2g}$$

In the stilling chamber (Figs. 15a and 15b), the velocity head at the point 1 is so small that $v_1^2/2g$ can be neglected without affecting the accuracy of the experiments. The formulas may then be simplified to

$$H_w = h \times 12.6 - \frac{v_2^2}{2g} \text{ (axial approach)}$$

$$H_w = h \times 12.6 + \frac{v_2^2}{2g} \text{ (tangential approach)}$$

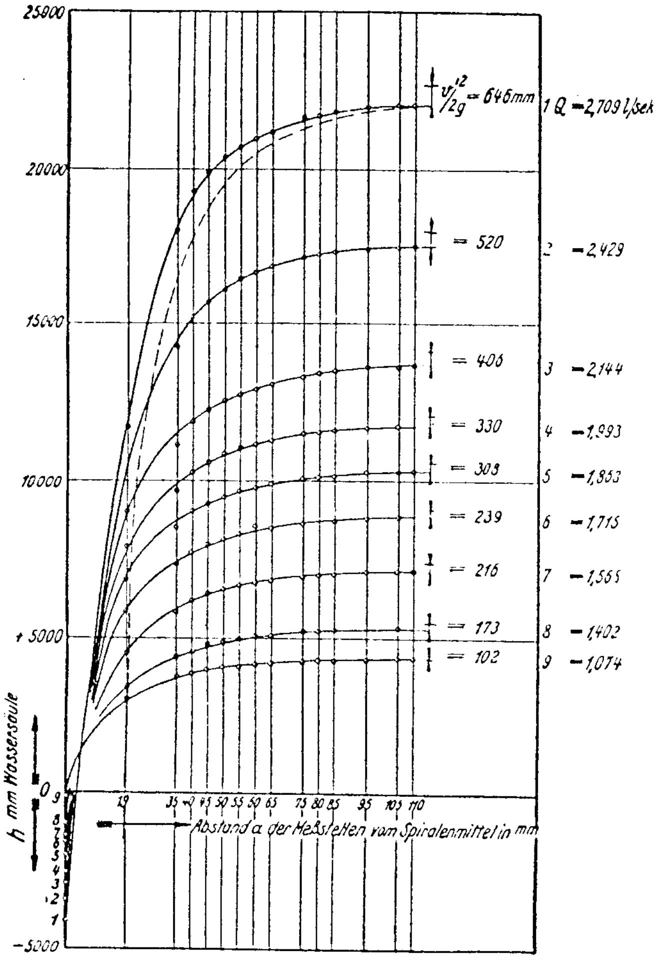

FIG. 18

(*Abstand a der Messstellen vom Spiralenmittel in mm.* = distance *a* from center of spiral vortex chamber in mm.; *Wassersäule* = head of water.)

The loss in the spiral casing is represented by curves in which it is plotted against the velocity head in the 38-mm. supply pipe $v_0^2/2g$. Approximately 15 different velocities were used in each experiment. The graphs permitted the correction or elimination of scattered points resulting from experimental or numerical errors.

For most of the experiments, H_w was proportional to $v^2/2g$, at least over a large range, and it is possible to represent the results of these experiments for both directions of flow by a coefficient of resistance,

$$\zeta = \frac{H_w}{v_0^2/2g}$$

where $\frac{v_0^2}{2g}$ = velocity head in a 38-mm. pipe.

From these values of ζ, a mean value was obtained and tabulated. The resistance coefficient for the direction of flow giving a high resistance will be indicated as ζ_1, and the value corresponding to the other direction by ζ_2.

The ratio of the resistance coefficients ζ_1/ζ_2 will be referred to as the "efficiency" in the following discussion.

SCOPE OF THE EXPERIMENTS

An explanation of the work and the results may be preceded by a brief summary of the experimental work.

1 Series I considered the variation of pressure within the larger vortex chamber.

2 Series II considered the relative efficiencies of the two vortex chambers as originally designed.

3 Series III and IV were conducted to clarify the effect of changes in the design of the tangential inlet.

Experiments I to IV used a large stilling chamber (Stilling Chamber No. 1), but in the following experiments the flow was straightened out by guides in a 40-mm. pipe (Stilling Chamber No. 2).

4 Alterations in the axial discharge were investigated, using the small vortex chamber (Series V). A cylindrical pipe was first placed so as to project different distances into the chamber, and then a nozzle-shaped tube, with various types of guide vanes (*Wirbelzerstörer*) was inserted in the axial discharge pipe (Series VI-IX).

5 Attempts to improve the operation of the valve by changing the shape of the vortex chamber were made, using the small vortex chamber (Series X).

6 Following the experiment with the smaller vortex chamber, the larger chamber was tested with the cylindrical pipe carrying the nozzle, fitted with guide vanes, projecting into the chamber (Series XI and XII).

7 A special experiment on the persistence of eddy motion in a long axial discharge pipe was also made (Series XIII).

THE PRESSURE DROP IN THE CHAMBER (SERIES I)

Pressure taps were made in the side cover of the large vortex chamber at different distances from the center (Fig. 17). These taps were attached to a manometer by means of a manifold, and the pressure at the various points could be measured by opening a pinch-cock. Before each experiment, the pressure differences across the spiral and the rate of flow were established. These experiments were conducted with nine different velocities of approach, and the results are represented in Fig. 18.

The abscissas are the distances *a* of the individual points of measurement from the center of the vortex chamber (Fig. 17)

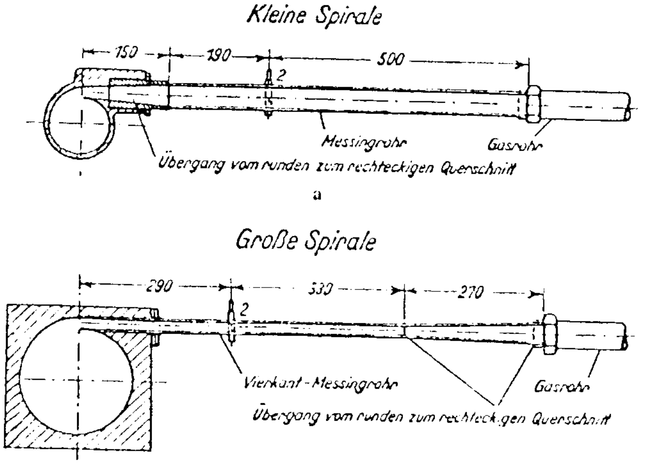
Kleine Spirale
150
190
500
2
Messingrohr
Gasrohr
Übergang vom runden zum rechteckigen Querschnitt
a
Große Spirale
290
330
270
2
Vierkant-Messingrohr
Gasrohr
Übergang vom runden zum rechteckigen Querschnitt
b

FIG. 19

(*Kleine Spirale* = small spiral vortex chamber; *Übergang vom runden zum rechteckigen Querschnitt* = transition from circular to rectangular cross-section; *Messingrohr* = brass pipe; *vierkant* = rectangular; *Gasrohr* = gas pipe; *Grosse Spirale* = large spiral vortex chamber.)

in millimeters, and the ordinates are the pressures above atmospheric in millimeters of water. The quantity $(v')^2/2g$ is the velocity head at the entrance area (20 by 38 mm.); in addition, the rate of flow is shown on each curve. At the abscissa zero, the pressure at the center of the spiral, as measured by a special manometer, is given. The curves form a system of cubical hyperbolas, but in the vicinity of the outlet area, at a distance of 35 mm. from the center, a deviation from this law appears.

The pressure in the spiral, neglecting the effect of skin friction and secondary eddies, was computed on the assumption of constant angular momentum ($c_u r$ = constant). The constant in this equation was obtained from the average velocity at the inlet section, and the tangential component of the velocity was then computed for different points in the spiral. The computed pressure curve appears as a dotted line in Fig. 18. This curve falls below the actual curve, and the ratio of the theoretical value to the actual, in the region of the pressure measurements (35 to 75 mm. from the center), is 1:1.4 to 1:1.5.

In the opposite (vortex-free) direction of flow, the pressure difference between a series of pressure taps in the chamber and the tap indicated as 1 in Fig. 11, was measured. For this condition, the pressure drop, below that at 1, was practically constant from 40 mm. from the center to the walls of the chamber. In the vicinity of the tangential outlet, the pressure was somewhat less, due to the high velocity (Fig. 17, points 1 and 3), while the pressures at points 11 and 14, adjacent to the axial inlet, were slightly greater. The pressure drop between point 1 and the vortex chamber amounted to 62 per cent of the total drop between 1 and 2 for the range of head stated.

Resistance to Flow (Series II)

For these experiments, the mercury differential manometer was connected between points 1 and 2, as is shown in Figs. 11 and 12. In Fig. 19a and b, only the point 2 appears.

The results are given in the following table:

				Range of experiments in mm. of water			
				ζ_1		ζ_2	
Experiment	ζ_1	ζ_2	ζ_1/ζ_2	H_w	$v^2/2g$	H_w	$v^2/2g$
Large chamber	72.7	3.87	18.8	17,000	229	8000	2700
Small chamber	32.5	3.93	8.3	16,500	492	7000	1805

The values given under the heading "Range of Experiments" show to what head loss and velocity in the 38-mm. pipe the experiments were carried. It was found that the values of the coefficient ζ given in this table, especially those of ζ_2, are valid only so long as no cavitation occurs. This phenomenon makes itself especially apparent at the smallest section of the tangential nozzle for the direction of flow giving the least resistance and will be taken up more exactly later. The cavitation limit did not lie within the range of the experiments now being discussed.

It appears from the table that the smaller vortex chamber presents an appreciably smaller resistance to flow in the direction of greatest resistance than does the larger one; the values of the resistance coefficient ζ_2, it will be noted, differ but little for the two casings. The ratio of the resistance coefficients for the different directions of flow is greater for the larger casing and therefore more favorable; this is due only to the fact that there is a considerably greater value of the coefficient ζ_1 for the larger casing.

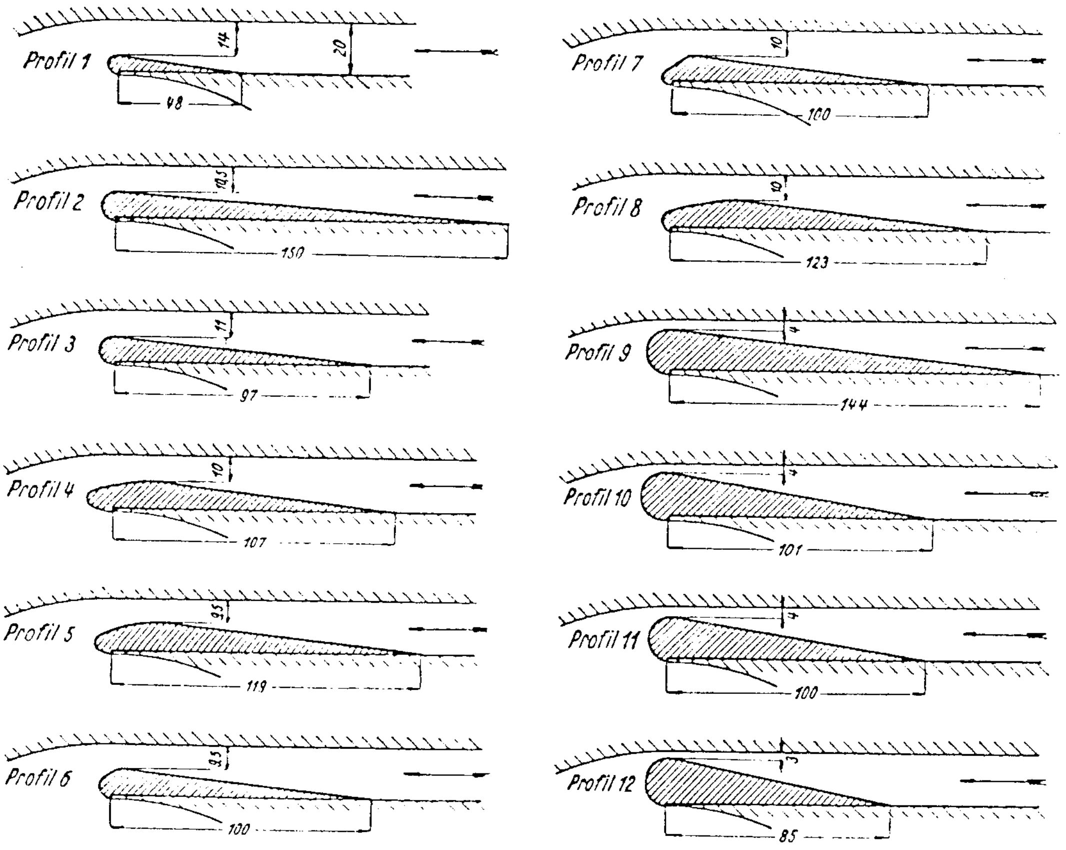

Fig. 20

(*Profil* = profile.)

Influence of the Shape of the Tangential Nozzle on Its Effectiveness as a Counterflow Brake (Series III and IV)

Alteration of the tapered throat between the tangential pipe and the chamber promised the greatest possibilities for improving the operation of the counterflow brake. For the large spiral, the throat piece was made of hard rubber, and for the small one, rounded metal pieces were used which at the same time served to contract the passage (Figs. 20 and 23).

It was to be expected that these alterations would increase ζ_1, the coefficient for the direction of greater resistance, since by constricting the tangential tube just before it enters the chamber, the flow is accelerated and the eddy is strengthened. Further, it was likely that, in the opposite direction of flow, a low pressure area would occur beyond the throat in the tangential pipe and increase the loss. It was hoped that the discharge capacity could be increased in the direction of small resistance by rounding the throat.

The experiments carried out in the larger chamber using the head control will be discussed next (Series III). The profiles tested are shown in Fig. 20, and the results appear in Fig. 21 and in the following table, where the numbers refer to the profiles.

The values of ζ_2 given in the table are valid only so long as cavitation does not occur. This phenomenon makes itself especially noticeable for the direction of small resistance at the narrowest portion of the throat; on the other hand, cavitation did not affect the value of ζ_1. To determine the value of ζ_2 without reference to the effect of cavitation, the regulation was accomplished by a valve beyond the chamber; thus the pressure head on the apparatus was equal to the static head of 17.09 m. diminished by the velocity head, $v_0/2g$, the measured loss H_w, and the friction loss in the supply line, which amounted to 7.1 $v_0^2/2g$.

SERIES III LARGE SPIRAL VORTEX CHAMBER WITH STILLING CHAMBER NO. 1

Profile	ζ_1	ζ_2	ζ_1/ζ_2	Range of experiments in mm. of water ζ_1 H_w	ζ_1 $v^2/2g$	ζ_2 H_w	ζ_2 $v^2/2g$	Clear width of throat piece in mm.	σ_1	σ_2
0	72.7	3.87	18.8	17,000	229	8,000	2700	20	<0.90	<2.6
1	95.0	4.31	22.0	14,700	150	4,000	940	14	<0.79	<4.1
2	119.3	5.06	23.6	15,000	124	11,400	1760	10.5	<0.80	<1.4
3	126.2	5.69	22.4	15,000	119	11,000	1244	11	<0.69	<5.6
4	115.9	5.40	21.5	15,000	134	9,000	1432	10	<0.70	<2.7
5	123.2	6.24	19.7	15,000	121	17,500	1750	9.5	<0.60	<3.3
6	129.4	8.91	14.5	17,000	132	17,000	1300	9.5	<0.65	<3.2
7	124.5	5.38	23.4	15,000	119	9,500	1300	10	<0.71	<4.0
8	105.0	5.43	19.3	16,500	154	16,000	1984	10	<1.03	<3.9
9	185.8	19.00	9.8	16,000	86	15,000	304	4	<0.75	4.0
10	210.7	31.83	6.6	16,000	74	14,500	202	4	<0.76	3.2
11	247.2	52.35	4.5	16,000	64	15,000	133	4	<0.76	3.5
12	297.9	95.83	3.1	16,000	53	16,000	75	3	<3.44	2.9

The pressure in the discharge is fixed by the head due to the head-control tank (17.09 m.), corrected for various friction losses in the supply line, the velocity head, and the loss in the check valve. The equation is:

$$\text{Pressure head in discharge line} = 17.09 - 8.1\frac{v_0^2}{2g} - \zeta\frac{v_0^2}{2g}$$

in which ζ is the resistance coefficient for either direction of flow.

A summary of the results indicated by the table follows:

1 A relatively slight constriction and streamlining of the throat, as in Profile 1, causes a great improvement in the efficiency. A greater constriction (Profile 2) did not futher improve the efficiency appreciably.

2 These improvements generally appeared for all profiles only for the direction of flow giving a vortex. Because of the constriction of the channel, a considerable increase in ζ_1 resulted for this direction of flow, but was not as great as had been expected. This is probably due to the braking action of the secondary eddies in the dead space separating the two streams of the main vortex as shown in Fig. 22.

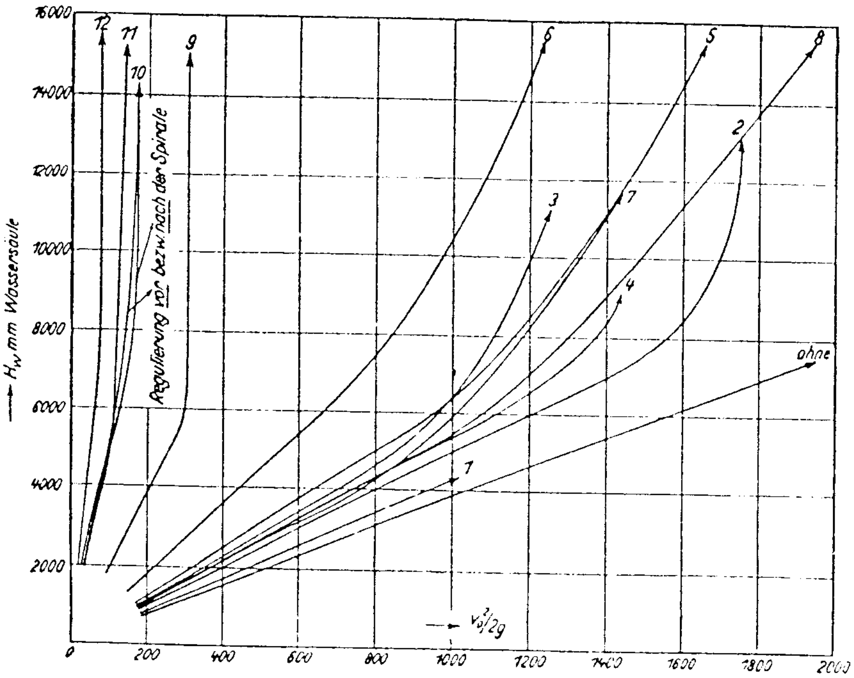

Fig. 21

(*Wassersäule* = head of water; *Regulierung vor bezw. nach der Spirale* = regulation by throttling in the supply pipe and in the discharge pipe, respectively.)

3 Rounding off the throat did not increase the discharge capacity for the vortex-free direction of flow. Values of ζ_2 for the profiles are in general higher than for the experiments without the throat pieces, showing that the constriction of the throat had an unfavorable effect in spite of the better guidance of the stream.

4 Sharpening the front face of the profile (Profiles 6 to 8) which was expected to give a better breaking away of the jet for tangential entrance, and flattening the front face (Profiles 4 and 5), which was expected to improve the discharge capacity for the vortex-free direction of flow, showed themselves to be unsuitable.

The value of σ_2 given in the table for Profiles 9 to 12 pertain to the cavitation limit which was reached even under the low head supplied by the head-control tank.

In a manner similar to what was done previously for water turbines,[3] D. Thoma has given the following derivation of the criterion for the danger of cavitation in the counterflow brake:

"As regards the danger of cavitation, the lowest pressure existing at any point is of the greatest importance. If we discharge various quantities of water through the apparatus, but choose the relations in such a manner that cavitation does not occur, then, as a result of the mechanical similarity of the flow, the pressure differences between any two points will be proportional to the squares of the rates of flow, neglecting the direct effect of the viscosity; this proportionality applies also to the difference between the pressure in the discharge line and the pressure at the (unknown) unfavorable location in the interior; the last-named pressure, which is the least occurring for the given condition of flow, may be indicated by $p_{\min.}$: it is then

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[3] Transactions of the First World Power Conference, London, 1924, paper no. 100.

$$\text{Discharge pressure} - p_{\min.} = k\frac{v^2}{2g}$$

where v is the mean velocity in the discharge line and k is an unknown coefficient associated with the type of construction.

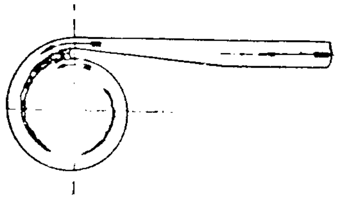


FIG. 22

Since the loss of head H_w is proportional to $v^2/2g$, the preceding equation can be written in the form

$$p_{\min.} = \text{discharge pressure} - \sigma\gamma H_w$$

where σ is associated with the type of construction; because of the mechanical similarity, the value of σ does not change for larger or smaller, geometrically similar designs, at least so long as the size or the velocity do not become so small or the viscosity so large that the direct effect of the viscosity is noticeable.

"To determine σ, an experiment was arranged in which the discharge pressure was so greatly reduced or the rate of flow so greatly increased that the first indications of cavitation were not quite evident. The value of σ is then obtained if we insert the measured value of the discharge pressure and the head loss H_w in the preceding equation and use for $p_{\min.}$ the vapor pressure of water.

"To determine whether cavitation is to be feared for any assumed application of the counterflow brake, compute $p_{\min.}$ by this equation; cavitation is not to be expected if $p_{\min.}$ is greater than the vapor pressure, that is, if

$$\text{Discharge pressure} - \sigma\gamma H_w > \text{vapor pressure}$$

"Since it is easier to express the pressure in the discharge in terms of pressure head, a slight transformation is desirable. Using H_a to indicate the barometric suction head,

$$H_a = \frac{\text{air pressure} - \text{vapor pressure}}{\text{unit weight of water}}$$

"If H_u denotes the pressure above atmospheric in the discharge line, the equation for the avoidance of cavitation becomes

$$\frac{H_a + H_u}{H_w} > \sigma$$

σ_1 will be used with flow in the direction which produces a vortex and σ_2 with flow in the direction which does not produce a vortex.

To determine the danger of cavitation for any type of construction, it is sufficient to carry out one experiment on the limit of cavitation for each direction of flow."

The values so obtained are given in the table, i.e., the values for the direction of small resistance as values of σ. For the other direction of flow the cavitation limit was never reached. With the arrangement for which the cavitation limit was not reached, the values given in the table are greater than σ for cavitation.

For the Profiles 1 to 8, the cavitation limit could be reached only by using the city water pressure, and under this condition, exact values of σ could not be computed because of the variation in pressure, which ranged from 4 to 5 atmospheres.

The initial experiments using only the pressure from the head control were repeated with the city water pressure and will be described here. The results obtained from these experiments for the direction of least resistance can be represented only by a graph, since their more or less irregular behavior at different velocities, the resistance for different throat profiles, and the point and type of deviation from the initially constant values of ζ_2 do not lend themselves to a tabular representation.

In Fig. 21, the ordinates are the loss of head H_w, and the abscissas are the velocity heads in the 38-mm. pipe. As in the previous experiments on Profiles 9 and 10, using the head control, the remaining values began to bend away from the curve. Up to a certain velocity head, there is a linear relation between $v^2/2g$ and H_w. On increasing $v^2/2g$ above a certain value, which lay between 280 mm. (Profile 9) and 1500 mm. (Profile 2), the values of ζ_2 given in the tables are no longer applicable. It should be noted that the increase in H_w was obtained by opening the valve beyond the spiral. The deviation from a straight line is caused by cavitation; after the cavitation had been completely developed, which occurred only with the very constricted throats (Profiles 9 to 12), the rate of flow did not increase with further reductions in the discharge pressure, so that the H_w line becomes vertical.

Profile 10 was used for an experiment on regulating the flow at a point upstream from the spiral; the different characters of the curves show clearly that the start of cavitation depends on the pressure conditions in the tangential tube behind the narrowest section. The same value of σ_2 was obtained by regulating the flow above and below the vortex spiral.

Although the experiments with the city water pressure did not make possible a determination of the value of σ_2 for Profiles 1 to 8, they gave, however, a basis for judging the operation of the various profiles at higher velocities and the possibility of setting a limit for the values of σ_2. Thus, Profiles 2 and 7 gave the same efficiency, but Profile 2 is shown by these experiments as the better one, since the possibility of cavitation in the direction of least resistance occurs considerably later than with Profile 7; this appears from the limits of σ_2 given in the table on page 21. Profile 2 gives a relatively greater discharge capacity in the direction of least resistance and a good efficiency (ζ_1/ζ_2 = 23.6 approximately).

In a manner similar to the procedure for the large vortex chamber, the tangential orifice of the smaller chamber was altered so as to have the different forms given in Fig. 23 (Series IV); Profile 0 is the one used in all previous experiments; Profiles 0 and 1 have an area of 20 by 38 mm. at the most constricted section, while the others have an area of 10 by 38 mm.

In representing the results for the smaller vortex chamber, a graph is unnecessary, since even on using the city water pressure no irregularities or deviations from the constant value of ζ_2 appeared. The value of σ is smaller for all profiles than the values given in the table on the following page.

The values of ζ_1 given in the table are valid over the whole range of the experiment, except that for the simple, sharpened throat (Profile 0, Nozzle 1) the values of ζ_1 were a little greater than appear in the table for H_w = 13,000; the same is true of Profile 1 for H_w = 8,500. However, these shapes are not important because of their low efficiency.

Profile 1 gave an appreciable reduction in the value of ζ_2 but simultaneously a greater reduction in ζ_1; apparently, the stream followed the bevel and the velocity decreased. The values of ζ_1 for Profiles 2, 3, and 4 are almost equal, but the values of ζ_2 show a greatly different discharge capacity in the direction of flow which does not give rise to a vortex. For this vortex chamber also, it appeared that a considerable contraction of the tangential supply pipe had a desirable effect on its operation as a counterflow brake, and a simple, semi-circular rounding gave the best results [Profile 2, ζ_1/ζ_2 = 16 (approx.)]. In this case, as well, rounding the throat piece increased the discharge capacity over that with a sharp throat for the direction of flow which does not give rise to a vortex.

SMALL VORTEX CHAMBER WITH STILLING CHAMBER NO. 1

Profile	ζ_1	ζ_2	ζ_1/ζ_2	Range of experiments in mm. of water				σ_1	σ_2
				ζ_1		ζ_2			
				H_w	$v^2/2g$	H_w	$v^2/2g$		
0	32.5	3.93	8.3	16,000	477	7,000	1,804	<1.75	<4.0
1	25.3	2.47	10.2	11,000	418	2,500	1,025	<1.38	<7.0
2	47.8	2.98	16.0	13,000	272	3,000	1,015	<0.92	<5.7
3	49.6	3.75	13.2	13,000	263	3,000	803	<0.92	<5.1
4	48.9	3.50	14.0	14,000	289	3,000	861	<1.02	<5.5

Experiments With a Different Type of Stilling Device

The stilling chamber and its group of pipes are intended to deflect the eddying stream paths in the axial pipe into parallel paths and thus to make possible a correct measurement of the water pressure downstream from the spiral. This arrangement was satisfactory for a determination of the pressure difference across the apparatus for the two different directions of flow and for choosing the best form of the tangential supply pipe, but the use of such a stilling chamber is impracticable in practice because of its size. Consequently, the best design of the vortex chamber, developed in the previous tests, was tested once with a simpler arrangement for straightening out the flow.

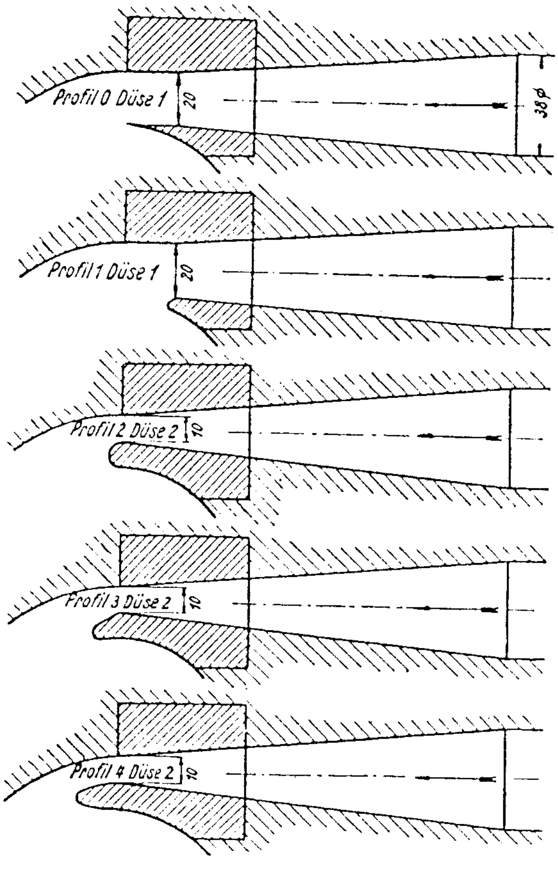


Fig. 23
(*Profil* = profile; *Düse* = nozzle.)

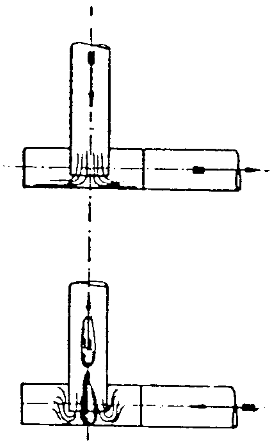


Fig. 24

Fig. 10 shows the experimental apparatus which replaced the large stilling chamber and which will be designated as Stilling Chamber No. 2. A suitable pipe 40 mm. in diameter and containing a group of small pipes 5 mm. in diameter and 50 mm. long was connected to the axial discharge piece and made watertight by stuffing boxes. This insured a deflection of the eddying motion into parallel paths. This group of pipes was later replaced by crossed metal plates of suitable length with the same effect, and in practice the latter arrangement appears simpler and more effective.

During the experiments, the mercury manometer was connected between points 1 and 2 as for Stilling Chamber No. 1 (Fig. 12). The pressure connections *a*, *b*, and *c* served only as a check on the pressure measured between 1 and 2. Between *b* and 1, the loss in the stilling chamber was measured separately for flow in the direction causing a vortex.

The following experiments, using Stilling Chamber No. 2, were carried out on both the large and small vortex chamber with the profile that had been found previously to be most suitable (Profile 2).

Before these experiments are described, a brief discussion of the influence of the alterations in the apparatus may be given. The following table compares the values of ζ for both vortex chambers with the different stilling devices but with vortex chambers otherwise unchanged.

	Stilling chamber	ζ_1	ζ_2	ζ_1/ζ_2
Large spiral with Profile 2	No. 1	119.3	5.06	23.6
and axial pipe at 0	No. 2	110.2	5.28	20.9
Small spiral with Profile 2	No. 1	47.8	2.98	16.0
and axial pipe at 0	No. 2	51.1	4.24	12.0

At this point, the resistance coefficient of Stilling Chamber No. 2 for the different directions of flow may be given. ζ_{r1} is the coefficient for vortex flow and ζ_{r2} the coefficient for vortex-free flow.

Resistance coefficient for Stilling Chamber No. 2	ζ_{r1}	ζ_{r2}
Large vortex chamber	12.0	1.54
Small vortex chamber	7.0	1.53

These values show the part which the stilling device has in the operation of the counterflow brake. Fig. 25b shows how the resistance of the Stilling Chamber No. 2 was measured.

EFFECT OF ALTERATIONS IN THE AXIAL DISCHARGE PIPE ON THE OPERATION OF THE SMALL VORTEX CHAMBER (SERIES V)

Since an important increase in the efficiency of the vortex chamber had been obtained by changing the form of the tangential supply piece, it appeared desirable to test the possibility of improvement in the design of the whole counterflow brake.

The consideration that an axial pipe which projects into the vortex chamber would not, up to a certain point, interfere with the entrance of water from the axial pipe (because it does not constrict the discharge area between pipe wall and side of chamber), but at discharge from the spiral into the axial pipe would strengthen the braking effect of the vortex and thereby must aid the desired operation, made it appear desirable to insert the axial pipe different distances into the chamber. The apparatus (Fig. 24) was so arranged that the axial discharge pipe could be inserted any distance into the chamber. It was desirable that, in the direction of flow giving a vortex, the rotating water should discharge as slowly as possible without causing a high resistance in the other direction of flow.

The following table shows the results of the series of experiments with the simple cylindrical discharge pipe (Series V).

SERIES V SMALL VORTEX CHAMBER WITH PROFILE 2 AND STILLING CHAMBER NO. 2

Setting of axial pipe, mm.	ζ_1	ζ_2	ζ_1/ζ_2	Range of experiments in mm. of water ζ_1 H_w	 $v^2/2g$	 ζ_2 H_w	 $v^2/2g$	σ_1	σ_2
0	51.1	4.24	12.0	15,000	271	4,000	947	0.78	<3.9
15	53.6	4.12	13.0	15,000	263	4,000	975	0.78	<3.8
20	55.8	4.30	13.0	15,000	250	4,000	936	0.71	<4.0
25	56.4	4.56	12.4	15,000	245	4,000	885	0.77	<3.7
30	58.4	7.40	7.9	15,000	237	6,000	815	0.70	<2.6

It is evident that by inserting the axial pipe 15 to 20 mm. an improvement in the efficiency is obtained. A further insertion decreased the ratio through a greater proportionate increase in the coefficient of resistance for the direction of flow which does not develop a vortex. Below the limit of σ_1 given, the value of ζ_1 was generally smaller, that is, the value was impaired by the appearance of cavitation.

As the next step in the improvement of the valve, the simple cylindrical axial pipe was replaced by a sort of nozzle, shown in Fig. 25a. This constriction of the cross-section of the axial pipe promised an improvement in ζ_1 for the vortex direction of flow, since the rotating water would be drawn closer to the axis, while in the opposite direction of flow, no important loss was to be expected through the pulling away of the stream from the walls in the expanding section. This nozzle was arranged so that it could be inserted into the spiral vortex chamber.

The results of the experiments with the nozzle in the small chamber and using Stilling Chamber No. 2 as in Fig. 10, are given in the following table (Series VI).

SERIES VI SMALL VORTEX CHAMBER WITH PROFILE 2, STILLING CHAMBER NO. 2, AND DISCHARGE NOZZLE WITHOUT GUIDE VANES

Setting of axial pipe, mm.	ζ_1	ζ_2	ζ_1/ζ_2	Range of experiments in mm. of water ζ_1 H_w	 $v^2/2g$	 ζ_2 H_w	 $v^2/2g$	σ_1	σ_2
0	112.3	5.77	19.5	14,000	134	5,000	878	1.30	<3.1
15	113.5	5.36	21.4	14,000	130	5,000	938	1.82	<3.3
20	118.5	5.13	23.1	14,000	129	5,000	975	1.82	<3.7
25	120.5	5.16	23.3	14,000	128	5,000	975	1.14	<3.5
30	121.5	5.83	20.9	14,000	126	5,000	863	1.12	<2.8

It appears from these experiments that the values of ζ_1 are more than double the values for the plain cylindrical pipe without a very great increase in ζ_2. The efficiency for a setting of 25 mm. reached 23.3. For values of σ_1 which are less than the values given in the table, scarcely any decrease in the value of ζ_1 due to cavitation was observed; that is, with the nozzle, the entrance of cavitation reduced the resistance of the apparatus for vortex flow.

The next step was the insertion of cross-pieces of sheet metal 2 mm. thick and 50 mm. long, sharpened at both ends, at and below the throat, to straighten out the flow (Fig. 25a). Later, in place of the cross-pieces, sharpened sheet metal plates, from 50 to 90 mm. in length, were inserted.

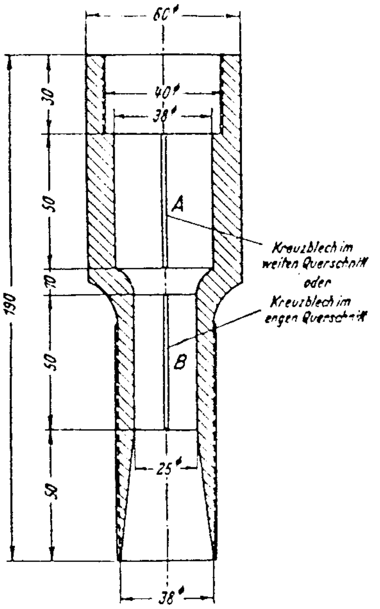

60ø
30
40ø
38ø
50
A
Kreuzblech im weiten Querschnitt oder Kreuzblech im engen Querschnitt
190
10
50
B
25ø
50
38ø

FIG. 25a

(*Kreuzblech im weiten Querschnitt* = crossed guide vanes at wide section; *Kreuzblech im engen Querschnitt* = crossed guide vanes at narrow section.)

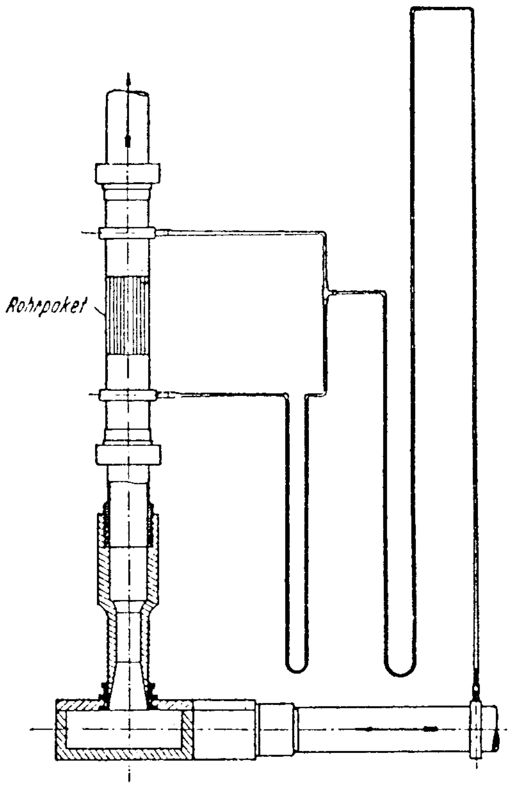


FIG. 25b

[*Rohrpaket* = Stilling Chamber No. 2 (literally, bundle of pipes).]

It is to be noted that for any guide vane in the discharge pipe, the resistance coefficient of the stilling chamber alone (ζ_{r1} for Stilling Chamber No. 2 in 40-mm. pipe), for the direction of high resistance, does not differ from the resistance coefficient for vortex-free flow. The resistance coefficients for the counterflow brake in the following tables refer to the stretch between the points b and 2 (Fig. 12). A complete straightening of the rotating water leaving the spiral was accomplished by the metal cross-pieces and the simple guide vane 90 mm. long.

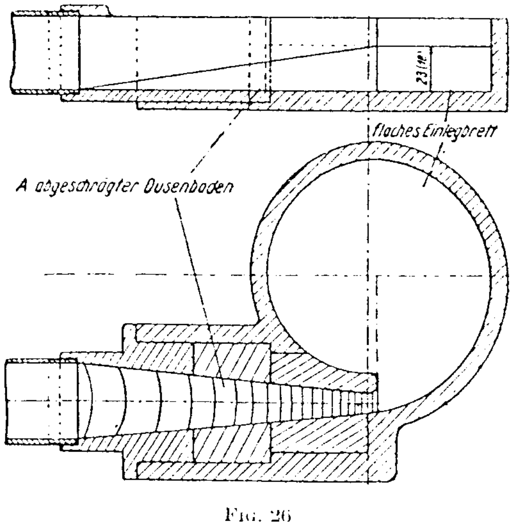


Fig. 26

(*abgeschrägter Dusenboden* = sloping base of nozzle; *flaches Einlegbrett* = flat plate inserted.)

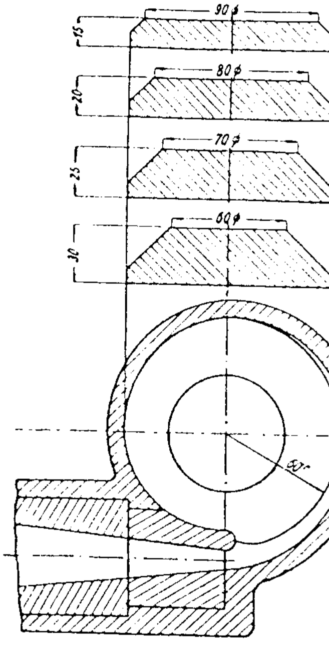


Fig. 27

The next table shows the results obtained with a nozzle containing a cross-piece inserted immediately beyond the narrowest section of the nozzle-shaped discharge pipe (Series VII).

In these experiments, the straightening of the flow occurred immediately behind the narrowest section of the discharge nozzle instead of in the stilling chamber. The difference in the efficiency, compared with that obtained in previous experiments, is in part to be explained by the fact that for flow in the direction of small resistance, the resistance of the cross-pieces is less than that of the group of pipes. Here also, for values of σ_1 less than those given, a decrease in ζ_1 occurs.

Since the crossed guide vanes actually straightened out the flow, the loss for flow with and without vortex formation was specially determined by means of the pressure taps above and below the stilling chamber (Fig. 25b), as was done for the values of ζ_{r1} and ζ_{r2} given on page 23. Since the water from the vortex chamber, after leaving the crossed guide vanes, loses as much pressure head in the stilling chamber as was lost for vortex-free flow, while the resistance of the same chamber without the crossed guide vanes is very much greater, it is permissible to assume that actually the vortex is destroyed in the guide vanes.

SERIES VII SMALL VORTEX CHAMBER WITH PROFILE 2 AND NOZZLE-SHAPED DISCHARGE PIPE CONTAINING CROSSED GUIDE VANE AT LARGER SECTION

Setting of axial pipe, mm.	ζ_1	ζ_2	ζ_1/ζ_2	Range of experiments in mm. of water ζ_1 H_w	ζ_1 $v^2/2g$	ζ_2 H_w	ζ_2 $v^2/2g$	σ_1	σ_2
0	120.2	4.07	29.6	15,000	137	5,000	900	1.12	<3.2
15	122.6	3.67	33.4	15,000	133	5,000	961	1.08	<3.4
20	124.4	3.79	32.8	15,000	133	5,000	944	1.05	<3.1
25	124.1	3.95	31.5	15,000	129	5,000	916	1.05	<3.3
30	124.3	4.24	29.3	15,000	129	5,000	870	1.05	<3.1

The use of crossed guide vanes at the narrowest cross-section appears undesirable from the data in the following table (Series VIII).

SERIES VIII SMALL VORTEX CHAMBER WITH PROFILE 2 AND DISCHARGE NOZZLE WITH CROSSED GUIDE VANES AT NARROWEST SECTION

Setting of axial pipe, mm.	ζ_1	ζ_2	ζ_1/ζ_2	Range of experiments in mm. of water ζ_1 H_w	ζ_1 $v^2/2g$	ζ_2 H_w	ζ_2 $v^2/2g$	σ_1	σ_2
0	94.9	4.90	19.4	15,000	158	5,000	791	<0.65	<2.8
15	96.5	4.59	21.0	15,000	155	5,000	834	<0.74	<2.9
20	97.1	4.99	19.5	15,000	152	5,000	791	<0.73	<2.8
25	99.3	4.99	19.9	15,000	149	5,000	791	<0.82	<2.8
30	101.1	5.55	18.2	15,000	146	5,000	713	<0.66	<2.7

The experiments conducted show that a nozzle of the type used is to be preferred to a simple cylindrical pipe, and that the desirable location of the guide vane is immediately downstream from, but not at, the narrowest section of the nozzle.

The next series of experiments considered the problem of replacing the crossed guide vanes (at A) with a sheet-metal plate. The results appear in the following table (Series IX).

SERIES IX SMALL VORTEX CHAMBER WITH PROFILE 2 AXIAL PIPE AT 0 MM.

Type of guide vanes at widest section of nozzle	ζ_1	ζ_2	ζ_1/ζ_2	Range of experiments in mm. of water ζ_1 H_w	ζ_1 $v^2/2g$	ζ_2 H_w	ζ_2 $v^2/2g$	σ_1	σ_2
Crossed guide vanes	120.2	4.07	29.6	15,000	137	5,000	901	1.12	<3.2
Sheet metal, plate, 50 mm.	126.4	5.71	22.1	15,000	161	5,000	892	1.12	<3.2
Sheet metal, plate, 90 mm.	124.9	4.22	29.6	15,000	129	5,000	875	1.12	<3.2

Using the method previously outlined, the experiments with the 50-mm. plate showed that the eddy was not completely destroyed (resistance with this arrangement, $\zeta_{r1} = 2.2$; with vortex-free flow, $\zeta_{r2} = 1.53$, page 23). However, the guide vane 90 mm. long completely destroyed the vortex, giving the same efficiency as the crossed guide vanes, and is to be preferred because of its simple construction. Here it must be pointed out that the value of ζ for the crossed and 90-mm. simple guide vanes does not contain the resistance coefficient of the stilling chamber in the pipe, while the value of ζ for the 50-mm. guide vane, which did not completely destroy the vortex, does include the resistance of the stilling chamber. This explains the great difference in the efficiencies given in the table.

INFLUENCE OF CHANGES IN THE VORTEX CHAMBER (SERIES X)

After these experiments, which were concerned with the possibilities of changing the tubes leading to the vortex chamber, it remained to test alterations in the chamber itself. In order to give the vortex chamber other dimensions, a hardwood plate was attached to the bottom of the spiral casing by screws. Two different kinds of plates were used, which will be referred to in the tables as "flat" and "conical."

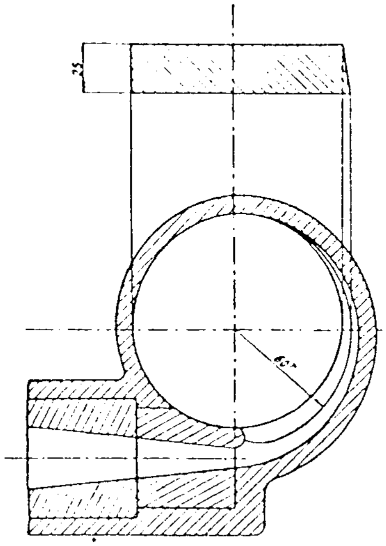


FIG. 28

The flat wooden plate covered the entire bottom. A smooth transition at the inlet was obtained by modeling with a kind of tar (Fig. 26 at A). The conical plate covered only a part of the bottom, and the tangential supply was not contracted (Fig. 27). These conical planks were made in different thicknesses, but all had a bevel of 45 degrees. For the most favorable thickness of plate obtained, a second experiment with equal thickness but much steeper angle (Fig. 28) was made, so that the tangential inlet remained the same, but the upper edge of the plate on the side of the throat piece coincided with the side of the spiral, similarly to the flat plate.

The results of this experiment appear in the following table and graph (Fig. 29).

For the experiments with nozzle inserted into the vortex chamber with and without the guide vanes (crossed guide vanes or simple vanes), a desirable decrease in value of ζ_2 was observed on insertion of the axial discharge line. However, contrary to expectation, the "impact-plate effect" did not occur with either the flat or the conical plate. With the flat plate, as the graph (Fig. 29) shows clearly, the heavy cavitation effect in the excessively constricted tangential inlet hardly appeared, but the conical plate appeared to impede the discharge of the water through this opening.

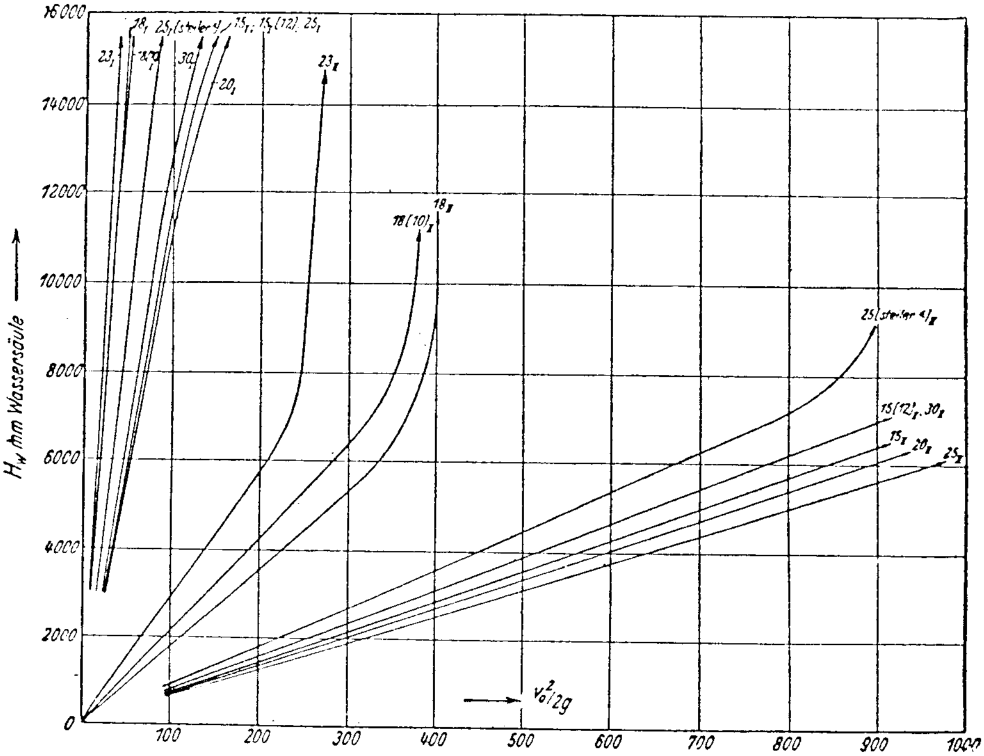


FIG. 29

(*Wassersäule* = head of water.)

The experiments with the plates led to no improvement in the efficiency.

SERIES X SMALL VORTEX CHAMBER WITH PROFILE 2 AND DISCHARGE NOZZLE WITH 90-MM. GUIDE VANE AT WIDE SECTION

Plate used		Setting of axial pipe, mm.	ζ_1	ζ_2	ζ_1/ζ_2	Range of experiments in mm. of water				σ_1	σ_2
						ζ_1		ζ_2			
Type	Thickness, mm.					H_w	$v^2/2g$	H_w	$v^2/2g$		
Flat	18 (Fig. 26)	0	320.9	16.25	19.7	16,000	50	11,000	400	<0.63	3.6
		10	317.5	19.69	16.1	16,000	54	11,000	379	<0.65	2.9
	23 (Fig. 26)	0	404.3	28.23	14.3	16,000	39	13,000	263	<0.62	3.2
Conical	15 (Fig. 27)	0	120.2	5.79	20.8	15,000	140	6,000	834	1.11	<2.6
		12	117.4	6.32	18.6	15,000	140	6,000	770	1.18	<2.3
	20 (Fig. 27)	0	112.4	5.25	21.4	15,000	152	5,500	815	1.27	<2.6
	25 (Fig. 27)	0	120.5	4.79	25.1	15,000	140	5,500	876	1.12	<2.8
	25 (Fig. 28)	0	185.6	[illegible]	24.9	16,000	91	9,000	892	<0.56	1.9
	30 (Fig. 27)	0	132.5	[illegible]	21.1	15,000	125	6,000	770	0.88	<2.5

The best efficiency of the small spiral (ζ_1/ζ_2 = 33.4, table on page 25) was obtained with the crossed guide vanes in the wide section of the nozzle, with unaltered spiral vortex chamber fitted with Profile 2, and with the axial discharge nozzle inserted 15 mm. However, the simpler 90-mm. guide vane with the discharge pipe flush with the wall of the chamber is to be preferred, since this arrangement gave the same efficiency (29.6) as that with crossed guide vanes and with the discharge pipe flush. (Compare first and last values in table on page 25.) On inserting the discharge nozzle about 15 mm., an efficiency of about 33.4 is to be expected for the simple 90-mm. guide vane.

EXPERIMENT WITH THE LARGE VORTEX CHAMBER (SERIES XI AND XII)

The procedure for the smaller spiral determined the program for the larger spiral.

The experiment was carried out with the cylindrical discharge pipe and with the discharge nozzle, but with the crossed guide vanes only at the best location; in this vortex chamber, also, a comparison could be made as to the degree to which the nozzle improves the efficiency as compared with that of the plain cylindrical pipe.

The following table gives the data on the larger vortex chamber with the cylindrical pipe (Series XI).

SERIES XI LARGE VORTEX CHAMBER WITH PROFILE 2, STILLING CHAMBER NO. 2, AND CYLINDRICAL DISCHARGE PIPE

Setting of axial pipe, mm.	ζ_1	ζ_2	ζ_1/ζ_2	Range of experiments in mm. of water ζ_1 H_w	 ζ_1 $v^2/2g$	 ζ_2 H_w	 ζ_2 $v^2/2g$	σ_1	σ_2
0	110.2	5.28	20.9	16,000	146	4000	766	<1.34	<2.8
15	112.4	5.60	20.1	16,000	143	5500	1005	<1.30	<2.5
20	110.9	5.23	19.7	16,000	145	5500	985	<1.33	<2.5
25	111.1	6.39	17.4	16,000	145	6500	1038	<1.33	<2.7
30	110.0	9.54	11.5	16,000	146	9000	953	<1.35	<1.1

As the following table of the experiments on the larger spiral with the discharge nozzle shows, the insertion of this tube into the center of the vortex appears to aid the discharge of water from the chamber; inserting the tube about 15 mm. gives a smaller value of ζ_1 than when it is flush (Series XII).

SERIES XII LARGE VORTEX CHAMBER WITH PROFILE 2 AND DISCHARGE NOZZLE CONTAINING 90-MM. GUIDE VANE AT WIDE SECTION

Setting of axial pipe, mm.	ζ_1	ζ_2	ζ_1/ζ_2	Range of experiments in mm. of water ζ_1 H_w	 ζ_1 $v^2/2g$	 ζ_2 H_w	 ζ_2 $v^2/2g$	σ_1	σ_2
0	273.5	5.49	43.3	16000	69	7000	1014	<0.59	<1.81
15	224.9	5.55	40.5	16500	74	7000	995	<0.68	<1.80
20	228.1	5.63	40.5	16500	72	7000	977	<0.66	<1.80

The value of the ratio ζ_1/ζ_2 of 43.3 is the best efficiency obtained in these experiments.

PERSISTANCE OF ROTATIONAL FLOW IN A LONG DISCHARGE TUBE (SERIES XIII)

Since there are circumstances under which it is difficult to insert a guide vane below the vortex chamber in the axial discharge pipe, it appeared desirable to investigate the persistance of rotational flow in the axial pipe.

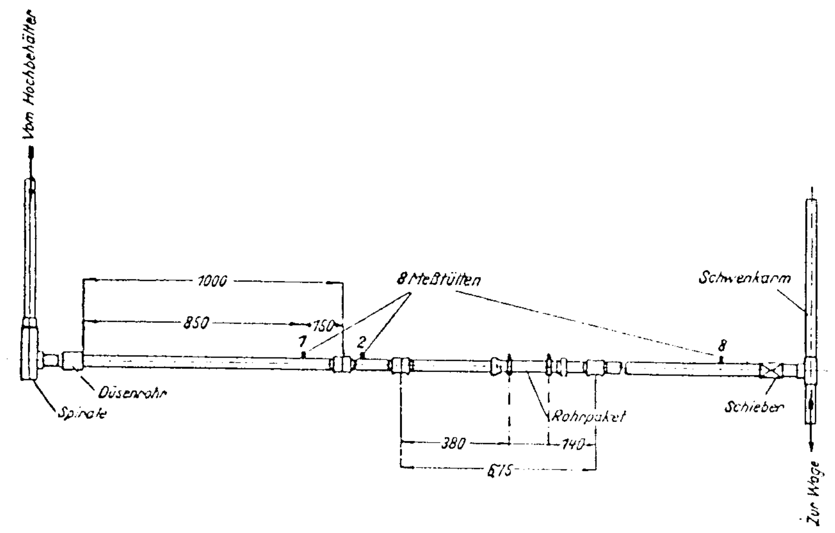


FIG. 30

(*vom Hochbehälter* = from head-control tank; *Spirale* = spiral vortex chamber; *Dusenrohr* = nozzle-shaped discharge pipe or discharge nozzle; *Rohrpaket* = Stilling Chamber No. 2; *Schieber* = valve; *zur Wage* = to scales; *Schwenkarm* = deflecting lever; *Messtüllen* = points of measurement.)

A pipe, 9 m. long, was attached to the smaller vortex chamber to determine the effect the chamber had on the type of flow in a long discharge pipe. This pipe line was made up of 8 one-meter lengths held together by flanges and bolts to make possible the insertion of Stilling Chamber No. 2 (Fig. 10) at different points in the line (Fig. 30). At these points, the loss was to be determined by measuring the pressure difference above and below the stilling device, which occurred because of the rotational component in the water leaving the vortex chamber. It was assumed that the smaller the tangential component, the lesser would be the deflection which the stream paths must experience and consequently the smaller the resistance. In this manner, by placing the stilling chamber at various points along the pipe and comparing the losses, an idea of the strength of the vortex at any point along the path could be obtained.

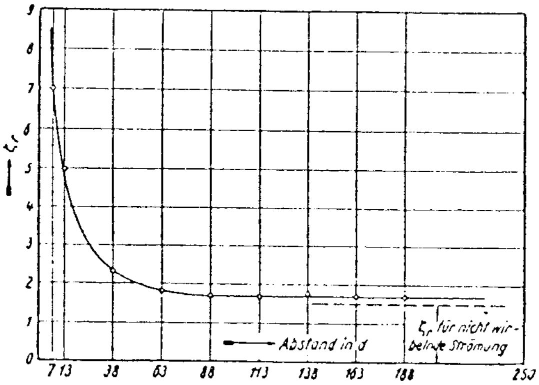


FIG. 31

(*Abstand in* d = distance in diameters; *für nicht wirbelnde Strömung* = for flow without vortex formation.)

The values of ζ_1 for the stilling chamber which were determined for different velocities at each location, remained constant except for very low velocities, for which the values of ζ_1 were higher.

The correlation between the resistance coefficient and the distance of the stilling chamber from the spiral is shown in Fig. 31. The distance is given in multiples of the diameter (40 mm.). The graph also shows the coefficient for eddy-free flow.

As was expected, the magnitude of the rotation decreased rapidly because of the braking effect of the skin friction. Beyond 83 diameters, no important change in the resistance occurs, but the resistance between 83 and 188 diameters differs somewhat from the value of ζ_2 measured for vortex-free flow, so that it must be assumed that the water still rotates.

Next, the drop in pressure along the length of the pipe was measured by pressure taps in each of the 1 m. lengths of pipe (Fig. 30). The pressure differences between each point and the ring-manifold connection in the stilling chamber at the end of the pipe were measured. The result of this experiment is shown in Fig. 32. The distances from the vortex chamber to the points of measurement are the abscissas, and the corresponding pressures are ordinates. The pressure drop for an equal mean velocity in vortex-free flow is shown as a dotted line for comparison. Beyond 90 diameters, the curves differ but little.

It is striking that beyond 90 diameters, the curve for vortex-free flow falls slightly below that for vortex flow, as is shown by the graph.

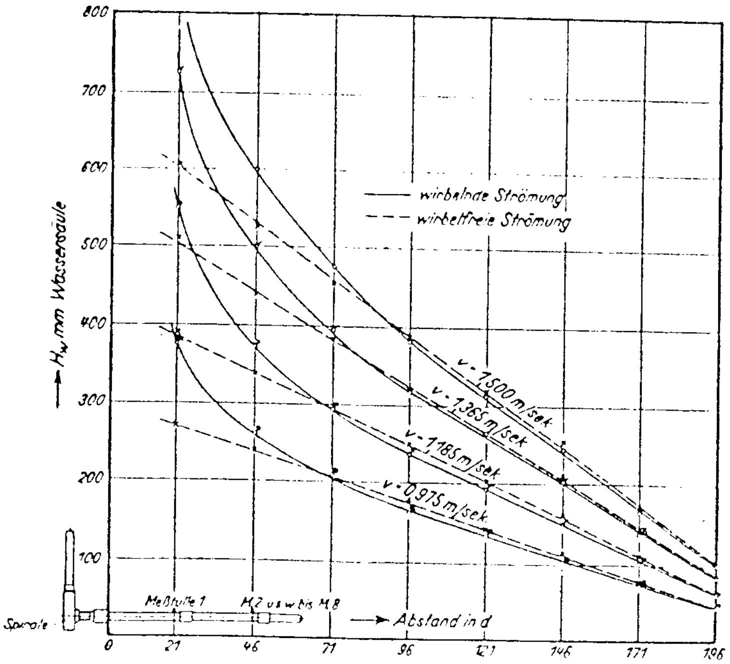


FIG. 32
(*wirbelnde Strömung* = flow with vortices; *wirbelfreie Strömung* = flow without vortices; *Messtelle* = point of measurement; *Abstand in* d = distance in diameters; *Wassersäule* = head of water.)

However, the arrangement of the apparatus was such that the accuracy of measurement was not very great.

We can conclude that at about 80 to 90 diameters beyond the vortex chamber, the pressure drop is about the same as for parallel flow, and that at this distance there is no objection, therefore, to discharging the water into any other apparatus, the vortex being no longer present, but the braking action of the counterflow brake will be somewhat reduced.

SUMMARY

If one desires to choose from the different forms of hydraulic counterflow brakes given in this work, the one having the highest possible efficiency without reference to the dimension of the apparatus, the best form is the spiral given on page 17 (Fig. 13) with a ratio of the radii of 10:12, a contracted tangential inlet (Profile 2), and a nozzle-shaped discharge pipe with simple guide vane immediately beyond the narrowest part of the axial discharge pipe.

If the occurrence of cavitation with very great pressure differences is to be avoided, the tangential nozzle indicated as 1 in Fig. 20 and in the table on page 21 is to be used. If it is not necessary to obtain the best efficiency but is important to keep the dimensions of the counterflow brake small, the smaller vortex chamber (Fig. 14), with a ratio of radii of 5 to 7, may be used. The tangential supply line should be equipped with Profile 2, given on page 23 and in Fig. 21; the axial discharge should be a nozzle with a simple guide vane at the same point as above, but the axial pipe should project inward about 40 per cent of the total height of the vortex chamber.

The results of this work can be summarized as follows:

We have succeeded in developing an apparatus, the "counterflow brake," which, with no moving parts, exhibits greatly different coefficients of resistance in the two directions of flow (highest efficiency, $\zeta_1/\zeta_2 = 43.3$).

A series of forms of the Thoma counterflow brake were investigated and the best type chosen. In practical application, care must be taken to avoid the region of cavitation.

Feb. 23, 1954

J. H. BERTIN ET AL

2,670,011

AERODYNAMIC VALVE

Filed Oct. 26, 1948

Fig. 1

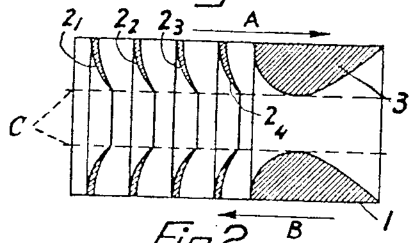


Fig. 2

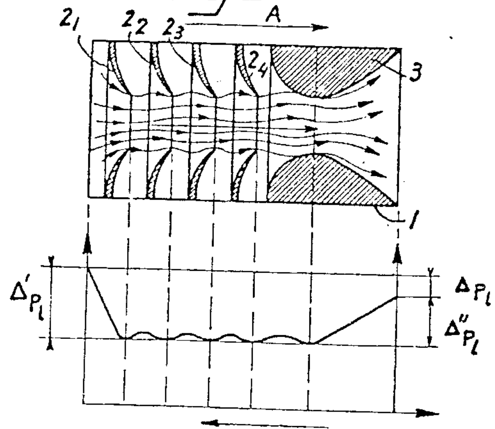


Fig. 3

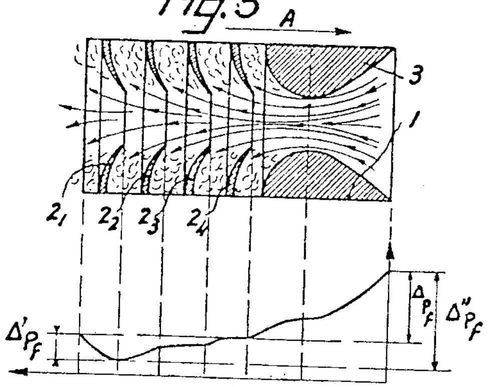
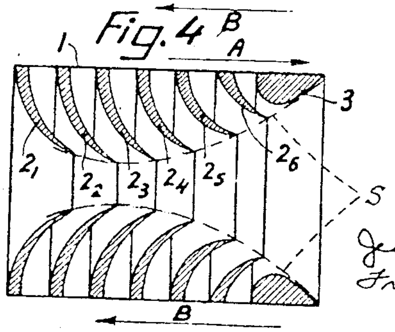


Fig. 4



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Patented Feb. 23, 1954

2,670,011

UNITED STATES PATENT OFFICE

2,670,011

AERODYNAMIC VALVE

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Application October 26, 1948, Serial No. 56,582

Claims priority, application France
October 31, 1947

5 Claims. (Cl. 138—44)

1

Our invention has for its object an arrangement providing a loss of head or a resistance for a gasiform stream that is small in one direction and considerable in the other direction, without its being necessary to resort to any movable mechanical members, this difference in action being provided by resorting to the laws governing the flow of gases so that the device may well be termed an aerodynamic valve.

This arrangement allows furthering the direction of flow in a predetermined direction inside a circuit of gases or vapors. When applied for instance to the exhaust of internal combustion or explosion engines, it allows improving the scavenging of the cylinders by preventing the rearward movement of the burnt gases, which is of particular interest whenever the exhaust gases are to feed a gas turbine as used for instance for driving a supercharger. But such an application mentioned merely by way of example should not be construed in a limiting manner and the apparatus may be used to advantage in all cases whereby reason of the fluctuations impressed to the progress of the gasiform stream, it is important to constrain said gases to advance in a single direction.

Our improved arrangement covers the association of a succession of nozzles all converging in the direction of flow that is to be furthered, with a diffusor mounted at the downstream end of said succession of nozzles in the direction considered. This diffusor may be constituted by a dissymmetrical Venturi-shaped member, the part of this member of which the free cross sectional area varies more slowly being located in the downstream side in the allowed direction of flow.

The input of fluid into the arrangement and its output may be provided along a common straight line which is an important advantage.

We have described hereinafter with reference to accompanying drawings given by way of example and by no means in a limiting sense an embodiment that will allow understanding how the invention may be properly executed, the features appearing both in the drawings and in the specification forming obviously part of the invention. In said drawings:

Fig. 1 illustrates in axial cross section a first embodiment of an arrangement according to our invention.

Fig. 2 is a diagram illustrating the flow of gases in the allowed direction with a graph showing the variations in pressure along the arrangement.

Fig. 3 is a similar diagram for the case where the flow is executed in the reverse or braked direction.

Fig. 4 is an axial cross section of a further embodiment of an arrangement according to our invention.

2

The embodiment illustrated in Fig. 1 includes a tubular section **1** that is to form part of a pipe system through which a pulsatory gasiform flow passes, a series of successive converging annular baffles forming discharge nozzles 2_1—2_2—2_3—2_4 and a Venturi-shaped member **3** beyond same. The direction of the allowed or free flow is that illustrated by the arrow A while the direction of the braked or hindered flow is illustrated by the arrow B and consequently the nozzles converge in the direction of the arrow A and the Venturi-shaped member **3** is arranged on the downstream side of the nozzles in the direction A; said member **3** is dissymmetrical and the portion thereof wherein the cross sectional area varies more rapidly is located foremost immediately beyond the downstream end of the nozzle system. The arrangement works in the following manner:

When the flow of fluid progresses in the direction of the arrow A, the first nozzle 2_1 has for its action merely to restrict the gasiform stream while reducing its pressure and increasing its speed. Each of the following nozzles 2_2—2_3—2_4 receives the stream from the preceding nozzles before its elementary streamlines, that are properly guided in the direction of flow, may diverge to any substantial extent, the spacing between the nozzles being suitably designed to this purpose. The speed and pressure of the gas in the stream, as considered in the first nozzle, remain substantially constant up to the entrance into the restricted portion of the Venturi-shaped member **3**. Beyond said portion the gasiform stream enters the downstream end of the member **3**, the cross section of which varies more slowly than in the input end. Consequently the high speed of the stream when considered in the restricted portion of member **3** may be damped with a good efficiency inside the diffusor formed by the Venturi-shaped member for the direction of flow considered, the wall of which flares out to a small extent only. The pressure thus recovered $\Delta''pl$ (Fig. 2) is to be deducted from the loss of head $\Delta'pl$ to which the fluid is submitted as it enters the first nozzle and which corresponds to the minimum pressure required for ensuring the throughput through the smallest cross sectional area provided by the arrangement. The actual loss of load produced by the arrangement is finally

$$\Delta pl = \Delta' pl - \Delta'' pl$$

where Δpl is very much less than $\Delta' pl$.

In the direction of flow corresponding to the arrow A, the flow is thus executed under excellent conditions.

When, on the contrary, the gases have a tendency to flow backwards in the direction of the arrow B, they meet first the slightly tapering portion of the Venturi-shaped member **3**. At the restricted portion of the latter, the speed obtained is considerable and by reason of the very large angle of divergence of the second portion of the member **3** that the stream meets thereafter, the elementary streamlines of the stream open rapidly and, by reason of the shape of the nozzles, the sharp edges of which are directed towards the flow of gasiform fluid, said elementary streamlines cannot be taken up under good conditions by said nozzles. The flow is submitted to a further loss of head each time it passes through one of the successive nozzles.

At the moment at which the flow passes through the opening of the last nozzle 2_1, it moves with a certain speed that is damped and may as precedingly be transformed into pressure so as to reduce the loss of head it is submitted to. But in the case considered, it is necessary on the contrary that this loss of head may be as high as possible. We provide for this, by constraining the elementary streamlines to move away as far as possible from the walls so that no rational damping of the speed may be obtained. It is necessary to this purpose that $\Delta' pf$ (Fig. 3) may be as small as possible; this requires, generally speaking, that the slope of the nozzles **2** may be sufficiently rapid.

With an arrangement so designed the loss of head in the braked direction

$$\Delta pf=\Delta'' pf-\Delta' pf$$

is much higher (six to eight times) than that in the opposite allowed direction

$$\Delta pl=\Delta' pl-\Delta'' pl$$

The advantageous properties of the arrangement depend to a considerable amount on the distance between the nozzles **2**. A testing under permanent flow conditions and then under pulsatory conditions will allow defining for each case the optimum spacing between nozzles.

The apparatus illustrated in Fig. 4 includes the same members as in the case of Fig. 1: a channel **1** containing a series of convergent annular baffles forming discharge nozzles 2_1—2_2—2_3—2_4 and a Venturi-shaped member **3**. But the nozzle openings instead of all having the same cross sectional area have different opening areas and are arranged in a manner such that their shape as a whole may correspond to the outline of a dissymmetrical Venturi-shaped member. In other words, whereas in the embodiment of Fig. 1 the periphery of the openings of the nozzles and a part of the neck or narrowest zone of the venturi **3**, lie on a cylindrical envelope surface C (shown in dotted line), in the embodiment of Fig. 4, the envelope surface S (also shown in dotted line) has a convergent-divergent shape, the convergent portion being shorter than the divergent portion which merges with a divergent part of the venturi.

The direction of allowed flow is again that given out by the arrow A while the opposed braked direction is given out by the arrow B.

The operation is substantially the same, but the length of the apparatus is less which may be of interest for alternating operation, the gaseous mass submitted to reversal of speed being smaller.

Obviously the above disclosed embodiments have been illustrated by way of exemplification and it is possible to substitute for their constituent parts equivalent technical means, without unduly widening thereby the scope of the invention as defined in accompanying claims.

What we claim is:

1. A unidirectional flow duct for allowing free downstream flow and hindering upstream flow, comprising in combination a plurality of dish-shaped annular baffles annularly engaging the inner wall of said duct and being longitudinally spaced therealong, and forming coaxial, converging passages, the least cross-sectional area of said passages being at the downstream end thereof, downstream of said baffles, an annular Venturi member, inside said duct, forming a converging-diverging passage having a neck portion of least cross-sectional area and coaxial with the former-mentioned passages, the periphery of the downstream ends of said converging passages and the periphery of the neck portion of said converging-diverging passage lying on a smooth envelope surface.

2. A unidirectional flow duct according to claim 1, wherein the converging part of said Venturi member is substantially shorter than the diverging part thereof.

3. A unidirectional flow duct for allowing free downstream flow and hindering upstream flow, comprising in combination a plurality of dish-shaped annular baffles annularly engaging the inner wall of said duct and being longitudinally spaced therealong, and forming coaxial, converging passages, the least cross-sectional area of said passages being at the downstream end thereof, downstream of said baffles, an annular Venturi member, inside said duct, forming a converging-diverging passage having a neck portion of least cross-sectional area and coaxial with the former-mentioned passages, the periphery of the downstream ends of said converging passages and the periphery of the neck portion of said converging-diverging passage lying on a smooth, converging-diverging envelope surface blending, at one end thereof, with the converging surface of the farthest upstream baffle and, at the other end thereof, with the diverging surface of said Venturi member.

4. A unidirectional flow duct according to claim 3, wherein the converging part of said envelope surface is substantially shorter than the diverging part thereof.

5. A unidirectional flow duct according to claim 1, wherein said envelope surface is of cylindrical shape.

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Frank W. Paul

Corrigenda to papers in Section A

Paper A.1 by Professor F.W. Paul

Page 6, last line - Figure 16 should read Figure 11.

Page 8, Acknowledgements - greatfully should read gratefully.

Add the following figure on prototype diode characteristics

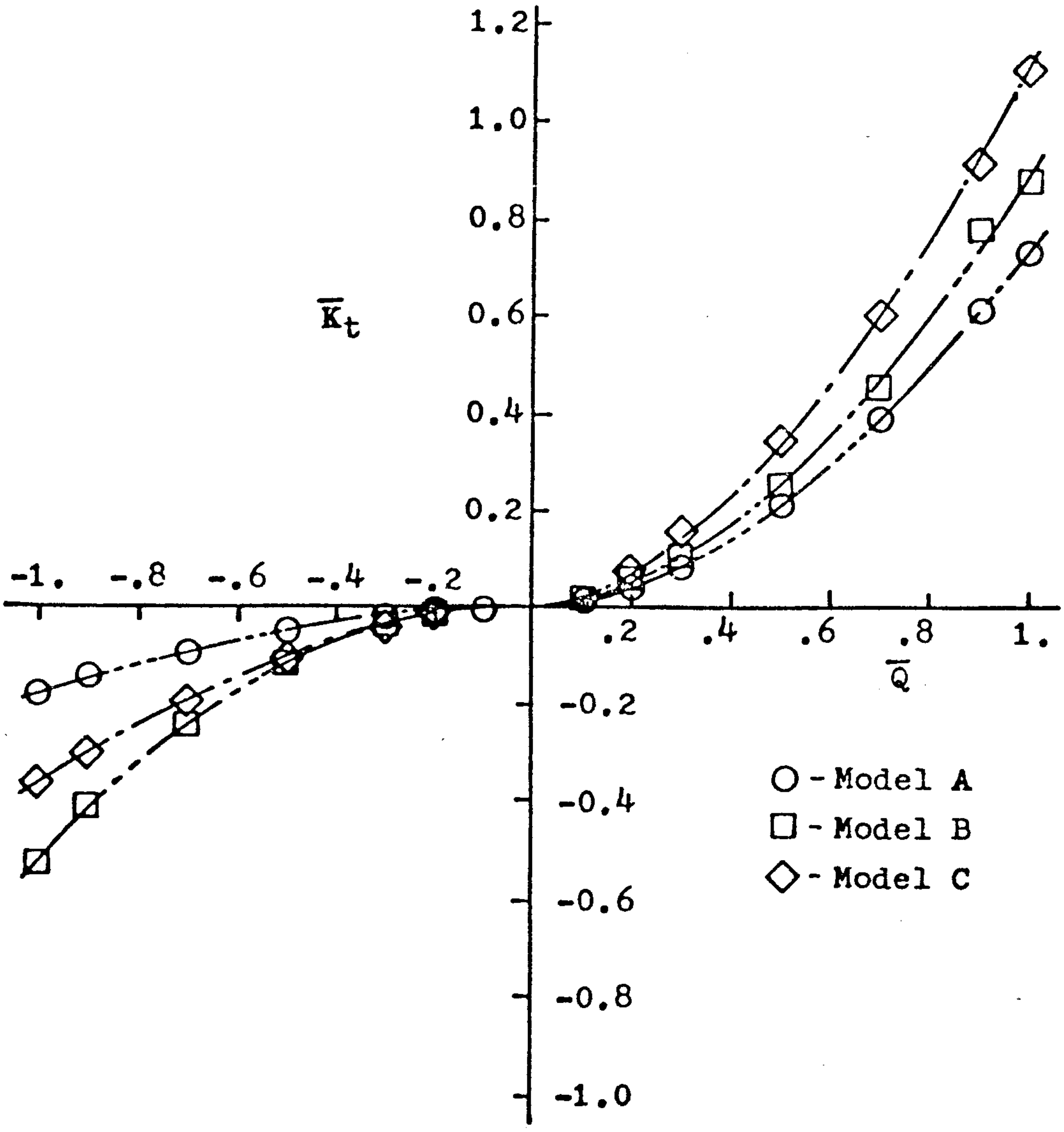

Figure 13. Total Pressure Loss Versus Diode Flow, Momentum Diodes.

FLUID MECHANICS OF THE MOMENTUM FLUERIC DIODE

Frank W. Paul

Introduction

The fluid mechanic characteristics of a one stage momentum or Tesla(1)* flueric diode are considered in this paper. The momentum diode is geometrically composed of a main channel which branches into two channels and are subsequently rejoined at a sharp angle. The internal geometry of the original Tesla diode is reproduced in Figure 1 (1).

Tesla claimed that "...A valvular conduit composed of a closed passage way having recesses in its walls so formed as to permit a fluid to pass freely through it in the direction of flow, but to subject it to rapid reversals of direction when impelled in an opposite sense and thereby interpose friction and mass resistance to the return passage of the same"(1). Tesla stated in his patent brief that the flow leakage thru each stage was represented by $(1/X)$. X was defined as the forward to reverse flow ratio. Tesla concluded that an n^{th} stage diode would permit only $(1/X)^n$ amount of total flow in the reverse to forward direction. The diode flow ratio X needed not be large to obtain a high flow ratio. Tesla also stated that the resistance of the reverse flow direction was 200 times that of the forward direction. Such simple statements overlooked many significant fluid mechanic principles.

The author has conducted a detailed study (2) on a one stage momentum flueric diode. Important considerations were (a) the optimum design for the steady-state diode resistance ratio and (b) the modeling and understanding of the basic fluid mechanic characteristics of the momentum diode.

Basic Fluid Mechanics of the Momentum Diode

All flueric diodes operate in two directions of flow, herein defined as reverse and forward. The total or static loss characteristics are considered when optimizing a diode for the maximum resistance ratio. This requires that any fluid diode be examined in the two modes of flow.

The momentum flueric diode utilizes confined jet interacting flows to provide the major pressure loss in the reverse flow direction. The secondary loss effects are flow separation

*Refers to the cited References

Frank W. Paul is presently Instructor, Mechanical Engineering, Lehigh U., Bethlehem, Pa., U.S.A.. Effective Sept. 1, 1968, Mr. Paul will be Asst. Professor, Mechanical Engineering, Carnegie-Mellon U., Pittsburgh, Pa., U.S.A..

at the downstream corners where the passages join and the turbulence loss occurring downstream of the jet intersection in the mixing flow region. These phenomena are indicated schematically in Figure 2(a) for the reverse mode of flow.

The forward loss characteristics are more difficult to define. One loss mechanism occurs in the boundary layer of the main channel. Another loss is caused by the side channel opening shown in Figure 2(b). Initial considerations of the problem indicate that flow passes from the main to side channel with some diffuser action.

A significant problem in optimizing a flueric diode for a maximum resistance ratio is to achieve simultaneously a high reverse loss coefficient and a small forward loss coefficient. This is not easily implemented. The work of the author suggests that the two modes of flow are inherently coupled by the geometry of the diode. Only by defining and recognizing all the fluid mechanic phenomena which occur, can the diode resistance ratio be maximized with respect to a given diode configuration.

The momentum diode has been studied by the postulation of analytic models with subsequent verification on a large scale geometric test model. These results were used to define the optimum or "best" momentum diode design.

Reverse Flow Model

The literature shows that few studies have investigated combining rectangular channel flows. Taylor(3), McNown(4), and Webber and Greated(5) treat combining flows with side channels, but do not develop general expressions for pressure loss characteristics.

The ideal model for reverse flow is one where no total pressure loss occurs between 0 and 1, and 0 and 2 of Figure 3. The model is idealized when the flow channel combination of streams 1 and 2 have the same total pressure. This is shown in Figure 4. The important geometric parameters of this model are the jet angle $\hat{\theta}$, the side to outlet channel widths λ , and the inlet to outlet mass flow ratio $\mathcal{X}$. Lambda and $\mathcal{X}$ are defined as

$$\lambda \equiv H_2/H_3 \quad ; \quad \mathcal{X} \equiv \dot{m}_1/\dot{m}_3 \; . \qquad (1)$$

Using a volume fixed in space and assuming that,

1. The flow is one-dimensional,
2. The flow is incompressible,
3. The fluid is perfect,
4. The flow is steady,
5. H_1 equals H_3,

the continuity and momentum equations can be applied. These equations in integral form are

$$\oint_s \rho \bar{V} \cdot d\bar{A} = 0 \qquad (2)$$

$$\oint_s \rho \bar{V}(\bar{V} \cdot d\bar{A}) = \Sigma \bar{F} \quad . \tag{3}$$

The continuity equation is written as

$$\dot{m}_1 + \dot{m}_2 = \dot{m}_3 \tag{4}$$

where the time derivative represents the mass rate of flow. The mass flow rates at each location are written as

$$\begin{aligned} \dot{m}_1 &= \rho H_3 \hbar V_1 \\ \dot{m}_2 &= \rho H_2 \hbar V_2 \\ \dot{m}_3 &= \rho H_3 \hbar V_3 \quad . \end{aligned} \tag{5}$$

The momentum equation is written in the X direction as

$$\dot{m}_3 V_3 - \dot{m}_1 V_1 + \dot{m}_2 V_2 \cos\hat{\theta} = H_3 \hbar (P_1 - P_3) \quad . \tag{6}$$

The static pressure loss from 1 to 3 is given by

$$P_1 - P_3 = \rho\left[\frac{H_2}{H_3} V_2^2 \cos\hat{\theta} + V_3^2 - V_1^2\right] . \tag{7}$$

The total pressure loss is given by

$$P_{t1} - P_{t3} = \rho\left[\frac{H_2}{H_3} V_2^2 \cos\hat{\theta} + \frac{1}{2}\left(V_3^2 - V_1^2\right)\right] , \tag{8}$$

where the total pressure is defined as

$$P_t = P + \rho V^2/2 \quad . \tag{9}$$

Equations (7) and (8) are rewritten in nondimensional form with respect to the average kinetic energy $\rho V_3^2/2$. The definitions of λ and χ are also introduced to give the static and total pressure loss coefficients for reverse flow.

$$K_R \equiv \frac{P_1 - P_3}{\frac{1}{2}\rho V_3^2} = 2\left[1 - \chi^2 + \frac{1}{\lambda}(1-\chi)^2 \cos\hat{\theta}\right] \tag{10}$$

$$K_{tR} \equiv \frac{P_{t1} - P_{t3}}{\frac{1}{2}\rho V_3^2} = \left[1 - \chi^2 + \frac{2}{\lambda}(1-\chi)^2 \cos\hat{\theta}\right] \tag{11}$$

A relationship between the mass flow ratio χ, channel ratio λ, and jet angle $\hat{\theta}$, permits the loss coefficients to be written in terms of the geometric parameters λ and $\hat{\theta}$. Two relationships based on fluid models are proposed.

The static pressure model considers that a stagnation region forms at the upstream corner of the combining flows. This is indicated in Figure 2(a). The static pressures at points 1 and 2 are then assumed equal as the stagnation region permits the pressure gradients across both channels to equalize. Assuming that the total and static pressures are equal gives

$$P_{t1} = P_{t2} \quad ; \quad P_1 = P_2 \tag{12}$$

then

$$\rho_1 V_1^2/2 = \rho_2 V_2^2/2 \tag{13}$$

or that

$$(V_1/V_2)^2 = 1 \tag{14}$$

for ρ a constant. Introducing the continuity equations and the definitions of $\mathcal{X}$ and λ gives

$$\mathcal{X} = \frac{1}{1+\lambda} \quad . \tag{15}$$

Equation (15) is independent of the jet angle $\hat{\theta}$.

Introducing Equation (15) into (10) and (11) gives expressions for the reverse loss coefficient for either static or total pressure loss in reverse flow.

$$K_R = 2\left[\frac{\lambda^2 + 2\lambda + \lambda\cos\hat{\theta}}{(1+\lambda)^2}\right] \tag{16}$$

$$K_{tR} = \left[\frac{\lambda^2 + 2\lambda + 2\lambda\cos\hat{\theta}}{(1+\lambda)^2}\right] \tag{17}$$

The complex potential model considers the flow as two-dimensional. The analysis uses complex variable theory and conformal mapping to obtain a functional relationship between $\mathcal{X}$, λ, and $\hat{\theta}$. This analysis is detailed in Reference (2). Only the resulting relationship is presented in this paper.

The resulting functional relationship is given by Equation (18).

$$(\mathcal{X})^{1-\frac{\pi}{\hat{\theta}}} - \lambda^{\frac{\pi}{\hat{\theta}}}(1-\mathcal{X})^{1-\frac{\pi}{\hat{\theta}}} = 1 \tag{18}$$

Equation (18) is solved in conjunction with Equations (10) and (11) to give the reverse loss coefficients as functions of λ and $\hat{\theta}$ using an iterative numerical scheme.

Figure 5 shows the theoretical reverse flow total loss coefficient as a function of λ and $\hat{\theta}$. The loss coefficient increases with increasing λ and decreasing $\hat{\theta}$. For values of λ greater than 1.5 the theoretical reverse loss coefficient is approximately constant. Only a 10 percent increase of the reverse pressure loss coefficient is obtained when the jet angle $\hat{\theta}$ is decreased from 45 to 0 degrees. Figure 6 shows the theoretical mass flow ratio χ as a function of λ and $\hat{\theta}$.

Forward Flow Model

The theoretical forward flow model is more difficult to formulate. The ideal model is a thru flow channel with no diffusing or side flow occurring in the branch channel. The thru flow channel model is not completely realistic for a real flow situation.

The best model comprises the loss due to the boundary layer and developing flow in the main channel plus the loss created by the side channel openings. Empirically the forward loss coefficient is written as

$$K_F \equiv \frac{P_3 - P_1}{\frac{1}{2}\rho V_3^2} = \Delta\bar{P}_{BL}\left(Re, \frac{h}{H_3}, \frac{\underline{L}}{H_3}\right) + \Delta\bar{P}_{BC}(\lambda) \tag{19}$$

and

$$K_{tF} \equiv \frac{P_{t3} - P_{t1}}{\frac{1}{2}\rho V_3^2} = \Delta\bar{P}_{tBL}\left(Re, \frac{h}{H_3}, \frac{\underline{L}}{H_3}\right) + \Delta\bar{P}_{tBC}(\lambda) \ . \tag{20}$$

The boundary layer loss $\Delta\bar{P}_{BL}$ depends on the Reynolds number Re, channel aspect ratio h/H_3, and channel length $\underline{L}/H_3$. The branch channel loss $\Delta\bar{P}_{BC}$ is a function of only the diode geometric parameter λ. The forward loss coefficient is evaluated on the experimental research facility.

Experimental Studies

A large model research facility was designed and built to verify the postulated reverse and forward flow momentum diode characteristics. Figure 7 shows a photograph of the assembled facility.

Tests were conducted on the research facility to substantiate the models for the reverse and forward loss coefficients. Two jet angle configurations of 45 and 90 degrees were studied. Various λ ratios were evaluated at both jet angles. The research model diodes were evaluated over a Reynolds number range of 70,000 to 170,000. The Reynolds number was defined using the average channel velocity and the main channel width.

Figure 8 shows the reverse total pressure loss coefficient for jet angles of 45 and 90 degrees. The agreement with the theoretical models is good. The static pressure model gives the better correlation for both jet angles. The total pressure loss coefficient is important since this parameter is used to correlate and compare the theoretical and prototype momentum flueric diode results.

Figure 9 compares the theoretical and experimental mass flow ratio χ with the geometric parameter λ for reverse flow. Results for jet angles of 45 and 90 degrees are shown in the figure.

Figure 10 shows the static or total pressure loss coefficients for forward flow. The loss for λ equal to zero is determined from rectangular channel losses which can be found in the literature (6,7). A limiting value for the forward pressure loss coefficient equal to approximately 0.26 is obtained experimentally.

Prototype Momentum Flueric Diodes

The model studies conducted on the research facility of Figure 7 represented the "best" flow configuration which could be made for a momentum flueric diode. The research model was designed to give the most favorable inlet flow conditions for reverse or forward flow operation. Prototype models required that some engineering compromises for the inlet diode flows be made.

Figure 11 shows the theoretical total diode ratio for the momentum flueric diode. The total diode ratio is defined as

$$R_t \equiv K_{tR} / K_{tF} \qquad (21)$$

The total diode ratio is shown for both the static pressure and complex potential flow models. The maximum diode ratios for jet angles of 45 and 90 degrees are 4.6:1 and 3.3:1 respectively.

The geometric parameters of jet angle $\hat{\theta}$ and channel ratio λ are considered in conjunction with the following items for a diode design.

1. The forward pressure loss is minimized for a given diode geometry.
2. The inlet reverse flow is divided to give the desired χ ratio for the reverse diode flow.
3. The limitations of the geometric design are of engineering practicality.

The minimization of the forward pressure loss coefficient severely limits the maximum attainable diode ratio. Since the diode ratio is defined as the reverse to forward total pressure loss ratio, a large reverse total pressure loss coefficient does not imply a large diode ratio. This paper reflects the opinion that attempts to increase the reverse total pressure loss coefficient by modifying the diode geometry can lead to increased forward pressure loss coefficients. The resulting diode ratio can then decrease.

The jet angle $\hat{\theta}$ and channel parameter λ are chosen in such a way to select the "best" or optimum geometry for a one stage diode. The predicted total diode ratio shown in Figure 16 is used to select the "best" diode design. The prototype diodes of this study have jet angles $\hat{\theta}$ equal to 45 degrees and a channel ratio λ equal to one. The diode ratio is constant for λ values larger than one.

Three prototype momentum diode models, shown in Figure 1 were constructed and experimentally evaluated in the laboratory. The average total diode ratios obtained for the prototype models are shown in Table 1. The diode Reynolds number varied from 1,700 to 17,000 for the experimental flow values.

Table 1

Model	Diode Ratio (total pressure)
A	4.07:1
B	2.22:1
C	3.18:1

A comparison of the average value of the total diode ratio with the theoretical prediction of Figure 11 shows that Model A gives a 10 percent smaller diode ratio. The lower experimental value is due to a smaller reverse flow loss coefficient. The theoretical total reverse pressure loss coefficient for λ equal to one is equal to 1.10 shown in Figure 5. The experimental value for prototype Model A is 0.734. The smaller value is due to the incorrect mass flow ratio χ in the main channel.

Model B improves the reverse total pressure loss coefficient by providing the theoretical mass flow ratio χ equal to 0.50 for λ equal to one. The experimental value of χ for Model B is 0.51. The agreement with the theoretical mass flow ratio χ is good although the experimental reverse pressure loss coefficient for this configuration is 0.88. The forward total pressure loss coefficient is 0.53 for Model B, considerably larger than for Model A. The increased forward loss is due to the sudden expansion of the main channel between points 1 and 0, shown in Figure 12. The average total diode ratio of 2.22:1 is obtained for this configuration.

The geometry of Model C directs the incoming reverse flow into the side channel. The experimental reverse total pressure loss coefficient of 1.11 agrees with the theoretical value of 1.10. The forward total pressure loss coefficient of 0.36 is obtained for this model. The forward loss coefficient is larger than predicted on the research diode. The increase is due to the curved channel for forward flow.

Models A and C of the prototype momentum diodes are the geometric extremes. The theoretical study predicts to within 10 percent the expected total diode ratio for the Model A diode. Less exact agreement is obtained for Model C. The poorer agreement is due to the increased forward total pressure loss coefficient.

Conclusions

The study conducted on the momentum (Tesla) flueric diode indicates that fluid mechanic models can be postulated which describe the steady-state diode ratio as a function of geometry. The reverse flow characteristic is idealized as an interacting confined jet flow. Two models are posed to describe the mass flow ratio as functions of the geometric parameters λ and $\hat{\theta}$. Both models are found to agree with results of the research model.

The forward flow model is more difficult to formulate. An empirical model is suggested and found to describe the forward flow behavior.

Comparison of the prototype momentum diode performance with the theoretical prediction shows agreement. A theoretical value of total diode ratio is 4.58:1 for λ equal to one and $\hat{\theta}$ equal to 45 degrees. The best experimental value of total diode ratio obtained for the prototype momentum diode is 4.07:1. Agreement is considered good since the analytic fluid models are based on ideal inlet flow conditions. The advantage of applying known analytic fluid mechanics methods to flueric diodes to predict steady-state diode ratio is established.

Acknowledgments

The author would like to express his appreciation to Dr. Alan H. Stenning, Professor of Mechanical Engineering, Lehigh University for his guidance and encouragement. The financial sponsorship of this work by Lehigh University is greatfully acknowledged.

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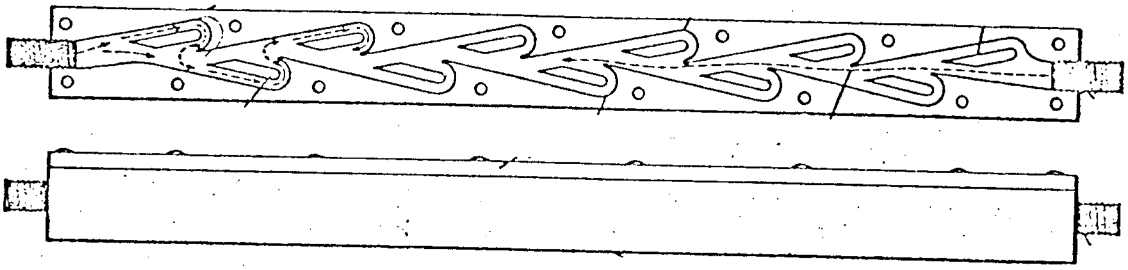
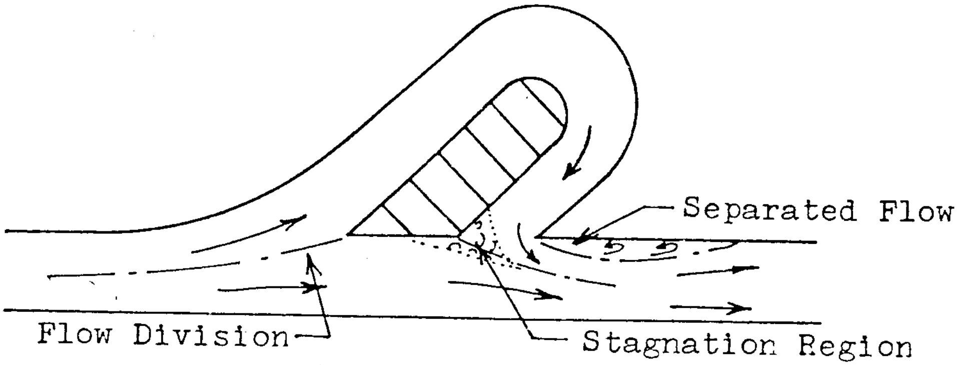
Nomenclature

$\bar{A}$	Area vector
$\bar{F}$	Force vector
h	Momentum diode depth
H_0, H_1, H_2, H_3	Momentum diode channel widths
K_R, K_{tR}	Nondimensional static and total reverse pressure loss coefficient
K_F, K_{tF}	Nondimensional static and total forward pressure loss coefficient
L	Momentum diode channel length
$\mathring{m}_1, \mathring{m}_2, \mathring{m}_3$	Channel mass flows
P	Static pressure
P_t	Total pressure
$\Delta P_{BC}, \Delta P_{tBC}$	Static and total forward pressure loss due to the side channel
$\Delta P_{BL}, \Delta P_{tBL}$	Static and total forward pressure loss due to the boundary layer
R_t	Steady-state total pressure diode ratio
$\bar{V}$	Velocity vector
$\hat{\theta}$	Jet angle
$\lambda = H_2/H_3$	Channel ratio
ρ	Fluid density
$\chi = \mathring{m}_1/\mathring{m}_3$	Reverse mass flow ratio

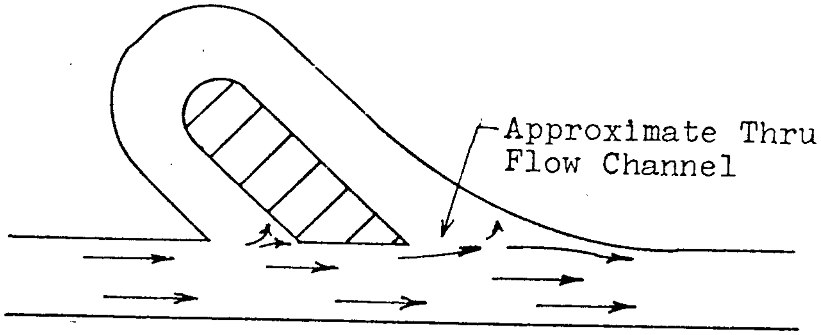


(Reproduced from Reference 1)

Figure 1 Tesla Diode



(a) Reverse Flow



(b) Forward Flow

Figure 2 Flow Mechanisms of the Momentum Diode

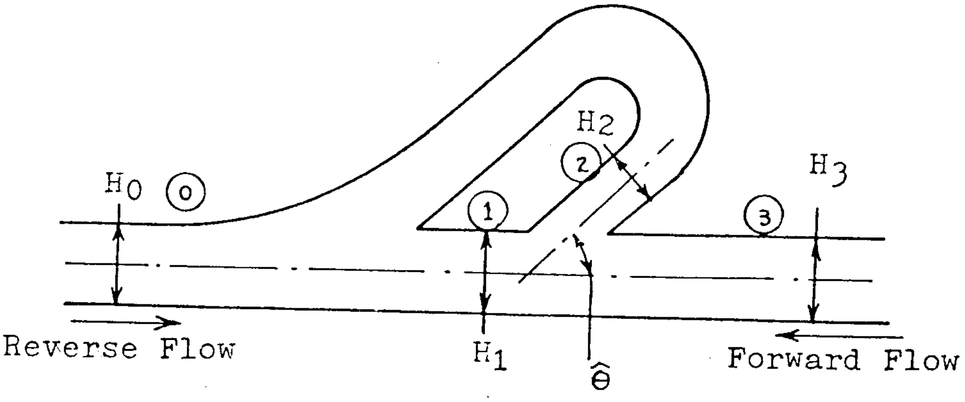


Figure 3 Momentum Diode Schematic

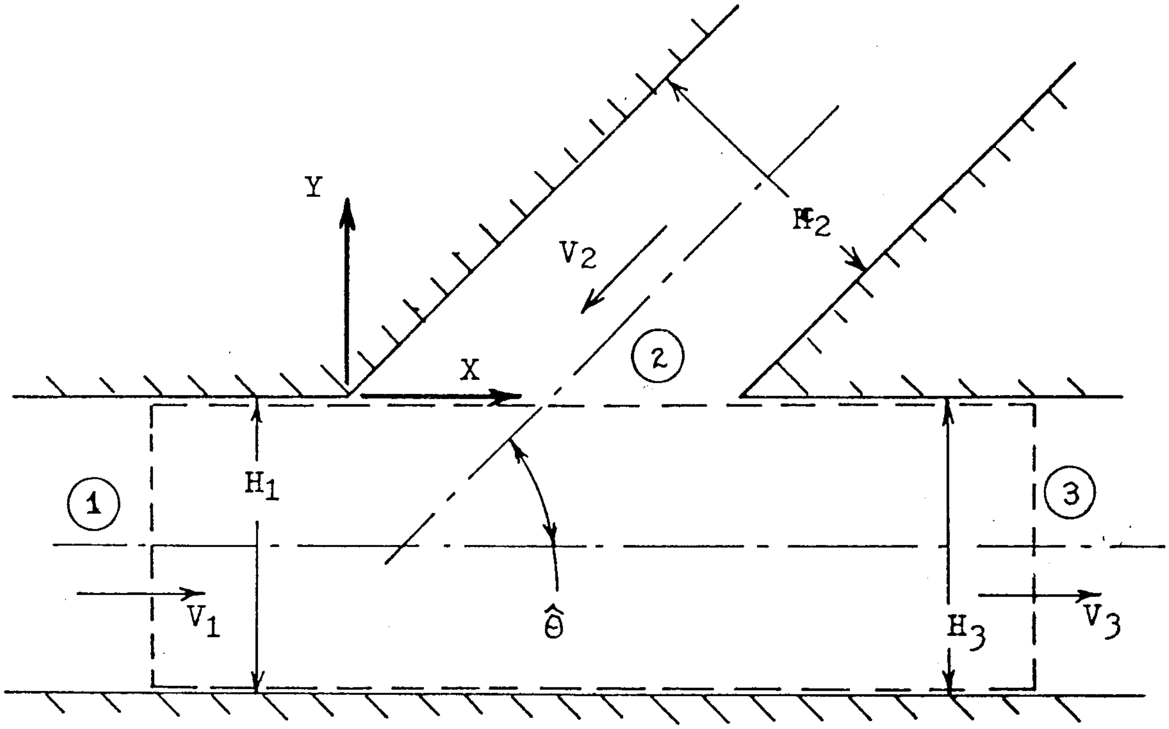


Figure 4 Ideal Reverse Flow Model, Momentum Diode

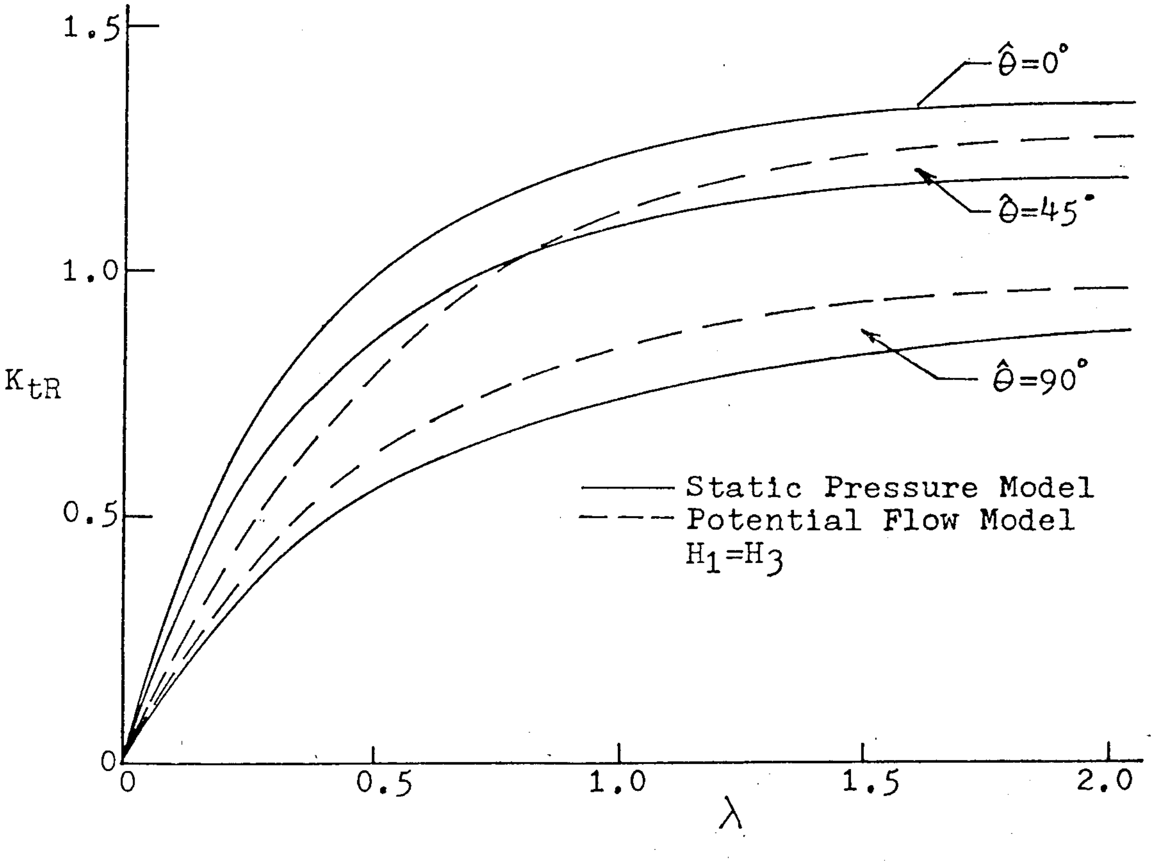


Figure 5 Theoretical Reverse Total Pressure Loss Coefficient, Momentum Diode

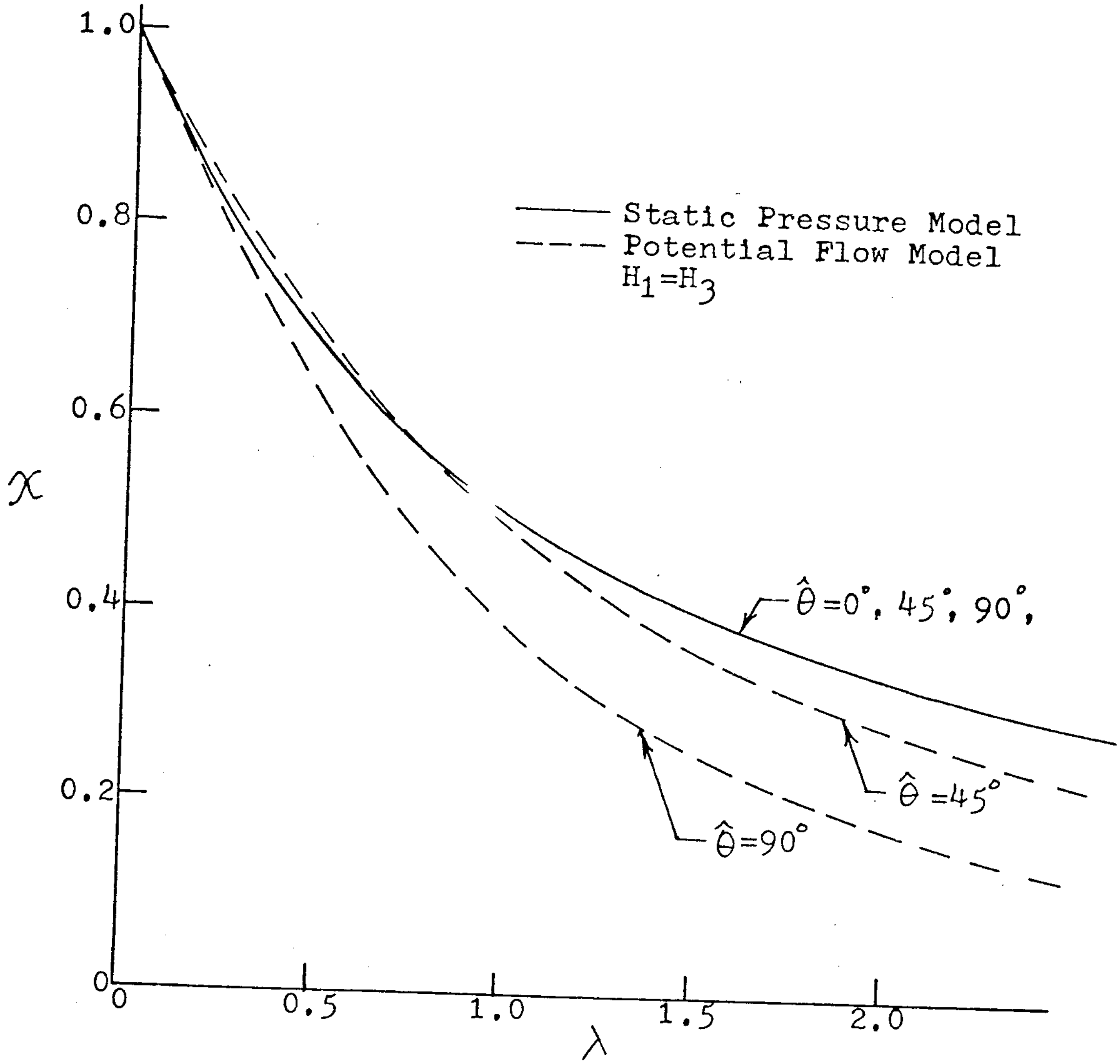


Figure 6 Theoretical Mass Flow Ratio χ, Momentum Diode

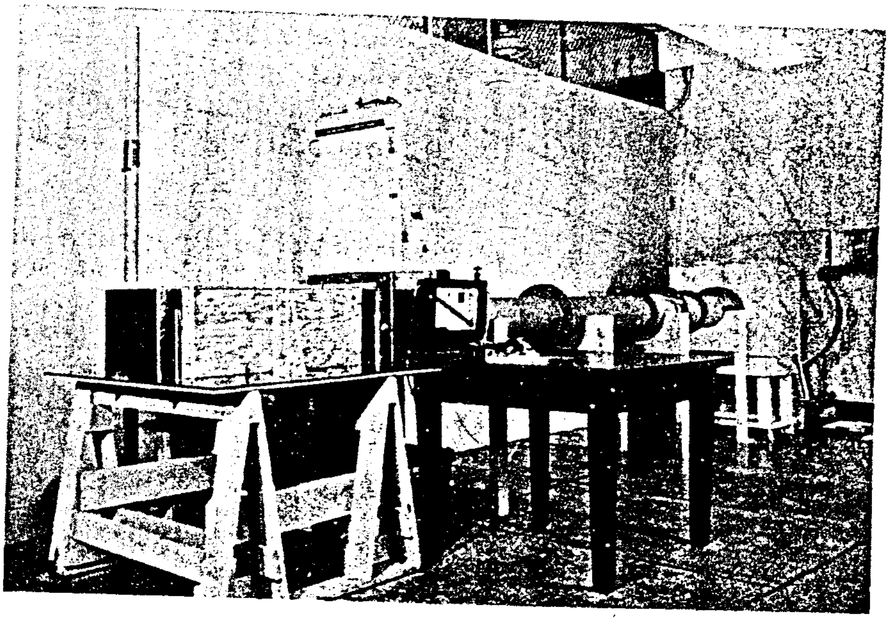


Figure 7 Research Diode Facility

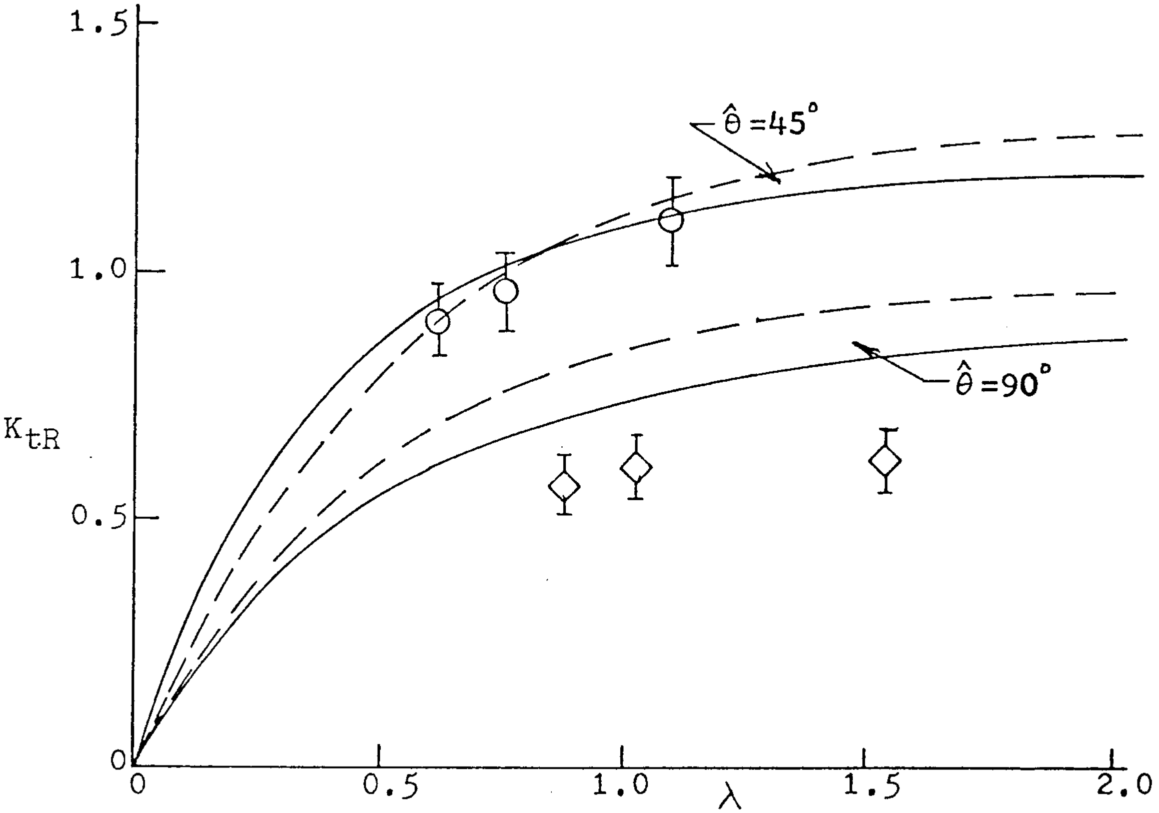


Figure 8 Experimental Reverse Loss Coefficient

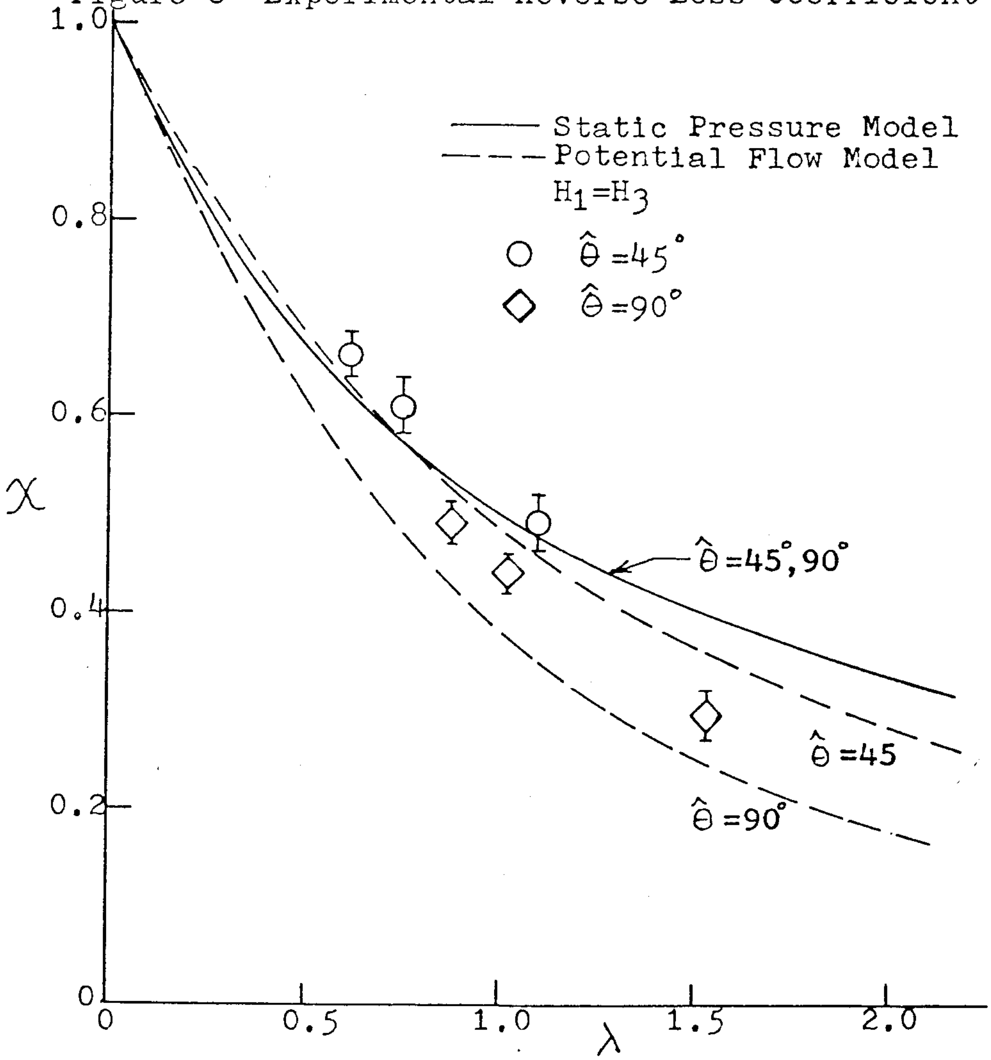


Figure 9 Experimental Mass Flow Ratio χ

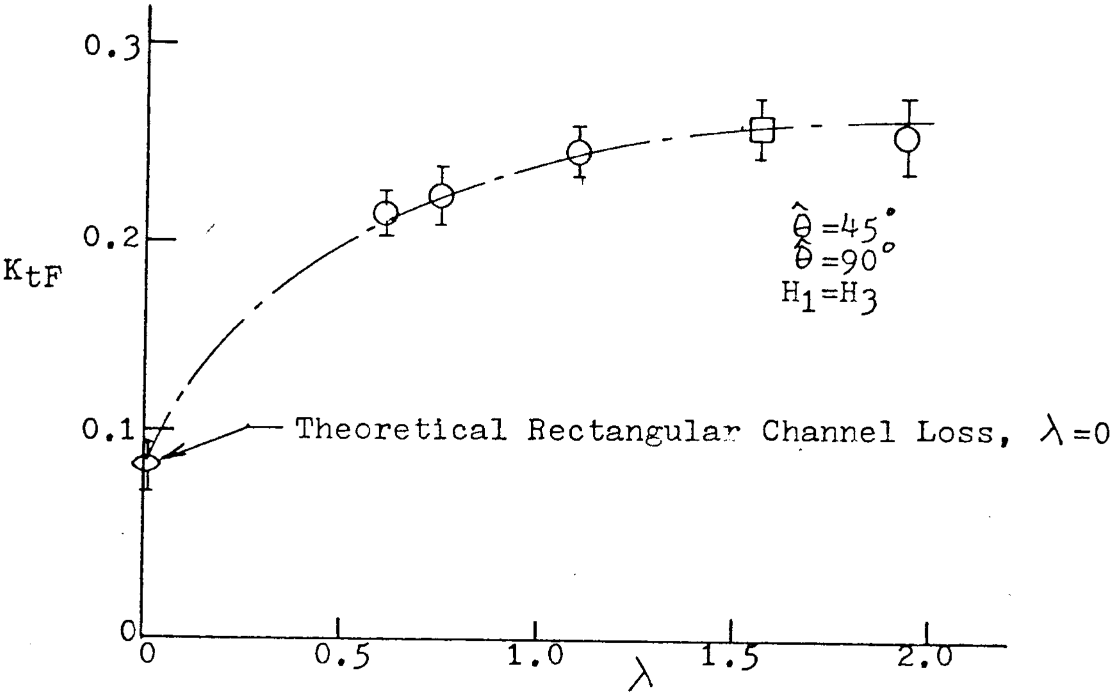


Figure 10 Experimental Forward Loss Coefficient

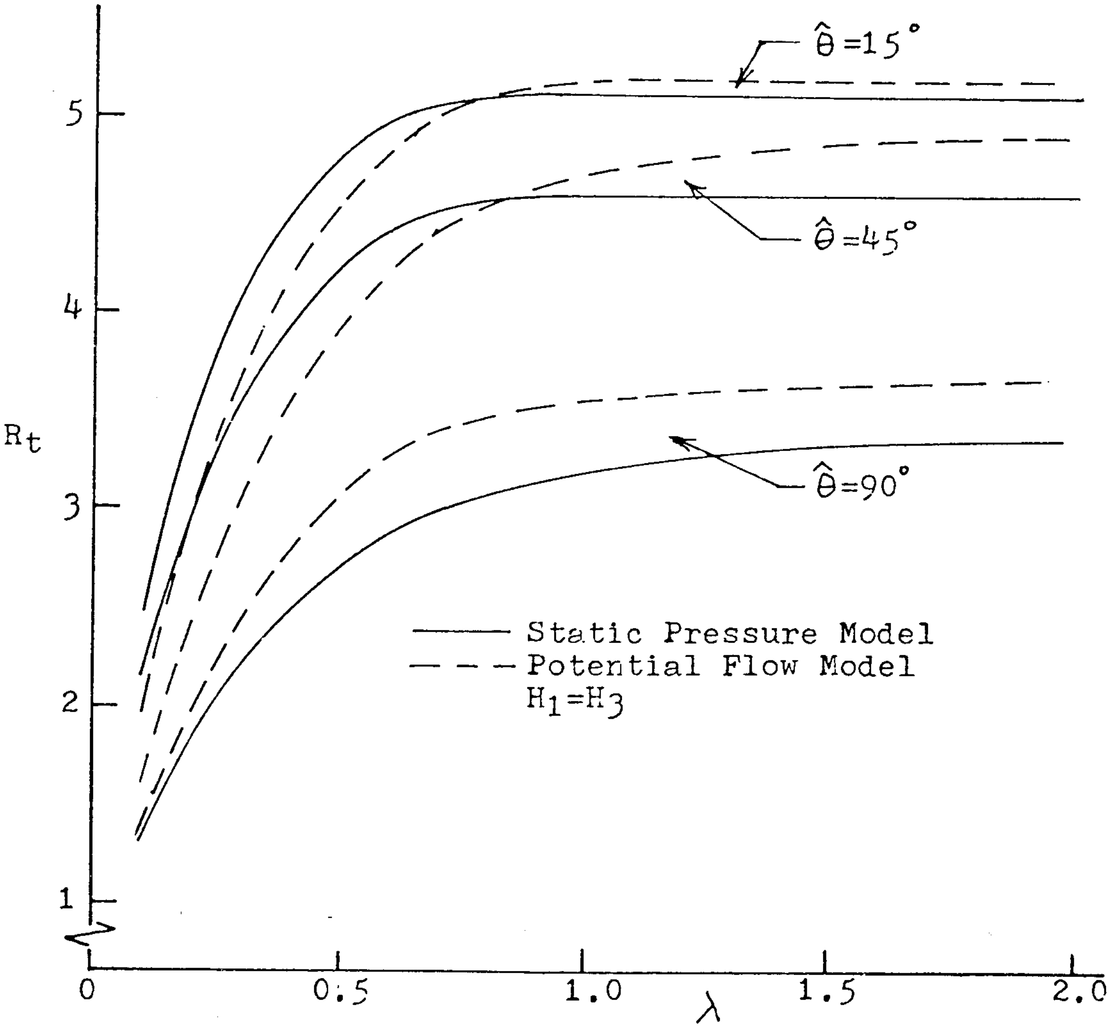
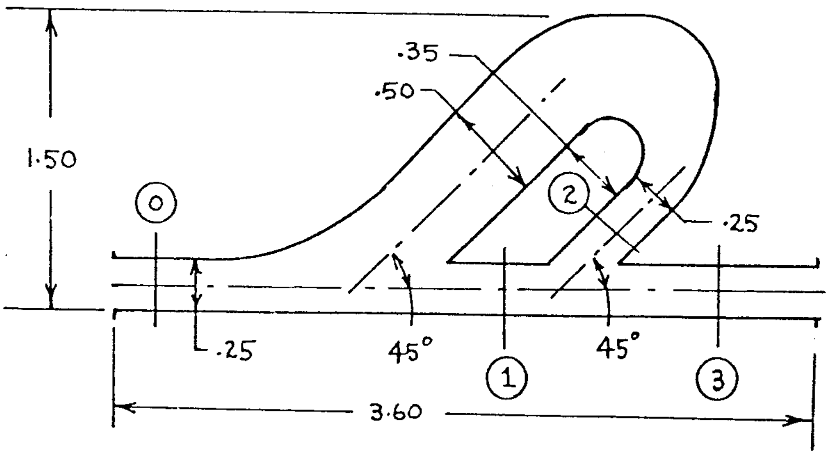
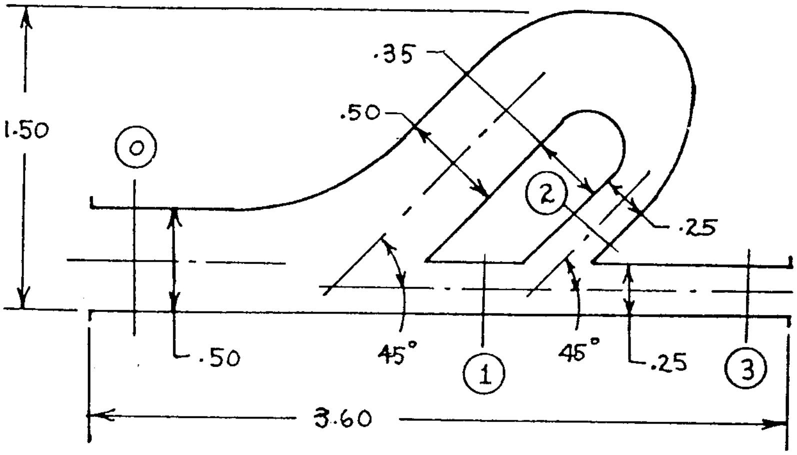


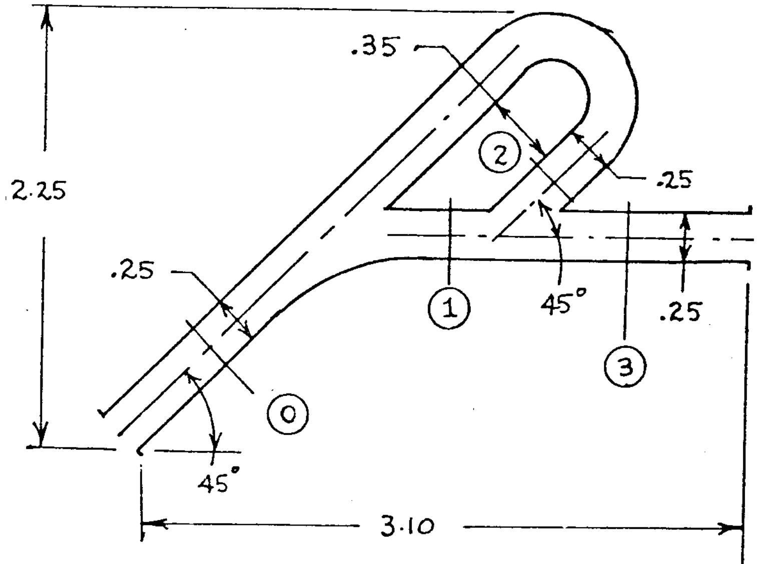
Figure 11 Total Diode Ratio



Model A



Model B



Model C

Channel Depth = 0.75 in.; Scale: Actual Size

Figure 12 Prototype Momentum Diodes

10b

THIRD CRANFIELD FLUIDICS CONFERENCE
8th - 10th MAY, 1968. TURIN.

PAPER K5

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p. K5-63 to K5-82

THE CASCADE DIODE

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The British Hydromechanics Research Association
Cranfield, Bedford, United Kingdom.

Summary

Experiments have been carried out on a cascade type of fluid diode in order to determine the ratio of the pressure drop in the reverse-flow direction to the pressure drop in the forward-flow direction at the same Reynolds number. This ratio, known as the performance, has been found to be strongly dependent on the length of the diode, over the range tested.

A simple analysis of the flow has been undertaken, and expressions for the reverse flow loss coefficient have been obtained.

DIODO A CASCATA

Sono stati compiuti degli esperimenti sopra un tipo di diodo fluidico a cascata per determinare il rapporto tra la caduta di pressione nel senso difficile e la caduta di pressione nella direzione del flusso facile, mantenendo costante il numero di Reynolds. Questo rapporto, che è un indice del comportamento del diodo, è stato trovato fortemente dipendente dalla lunghezza del diodo, nell'ambito delle prove eseguite.

E' stata compiuta una semplice analisi del flusso e sono stati ottenuti dei coefficienti di perdita per il flusso difficile.

Held at the Istituto di Tecnologia Meccanica del Politecnico di Torino, Italy
Printed and Published by the British Hydromechanics Research Association, Cranfield, Bedford, England.

Nomenclature

A	Cross-sectional area of diode
A_f	Area available for flow between cascades
a	Projected area of each blade
C_D	Drag coefficient of aerofoils
C_L	Lift coefficient of aerofoils
c	Blade chord
C_f	Friction Factor for flat plate
D	Diode diameter
d	Central bar diameter
F	Total force/blade in the direction of the diode centre line
f	Friction factor for pipe flow $(\Delta P = f \frac{\ell}{D} \rho \frac{U^2}{2g})$
g	Acceleration due to gravity
K_F	Forward flow loss coefficient
K_R	Reverse flow loss coefficient
K_{SF}	Loss coefficient due to shear forces on diode wall and central bar
ℓ	Length of diode
N	Number of blades per cascade
n	Number of cascades
ΔP	Pressure difference across diode
p	Blade pitch
Re	Reynolds number based on diode diameter
Re_f	Reynolds number based on length of flow path
S	Shear force on diode wall, and central bar
U	Mean velocity at diode entry
U_f	Mean axial component of velocity between blade cascades
V	Velocity of fluid approaching leading edge of blades
V_f	Mean velocity of fluid approaching leading edge of blades
β	Angle between flow direction and a perpendicular to the cascade axis
β_m	Mean flow angle
δ	Stagger angle of blades
θ	Helix angle measured over 15" length
μ	Absolute viscosity
ρ	Fluid density
Σ	Diode performance
τ	Wall shear stress

1. Introduction

The diode allows fluid to pass through it in one direction with a relatively low pressure drop, compared with the much higher pressure drop generated when the direction of flow is reversed. It is therefore possible to use these devices in circumstances in which it is required to limit the reverse flow in a fluid circuit. Such a reverse flow may be brought about by a surge. Clearly the fluid diode does not completely stop the flow as would a non-return valve. However, under adverse environmental conditions the use of a non-moving part device can be desirable.

The ratio of the flow rates in the forward to the reverse-flow directions, for a given pressure drop across the diode, is equal to the square root of the performance. Thus the experimentation was orientated towards obtaining the largest possible value of the performance. While other types of diode exist, previous work (1) has indicated that the cascade type (2) showed good possibilities for development.

2. Operating principle.

The diode, Figs 1 and 2, consists of several cascades of blades, (usually four), mounted on a central bar within a circular pipe. In the forward-flow, or low-loss direction the fluid tends to flow mainly between the cascades, although eddies are set up between the blades. In the reverse-flow, or high-loss direction the fluid is deflected by the blades and passed from one cascade to another. The flow deflections are associated with momentum changes in the axial direction which result in the relatively high resistance to flow.

The helical motion of flow is started more quickly by placing the blade cascades on a helix, and by setting the initial blades at progressively greater angles to the axis of the diode.

3. Experimental apparatus.

Testing of this diode has been carried out in three separate rigs. Two of which have already been described in a previous paper (1). The requirements of this diode, in terms of pressure and flow rate, were too great for the existing apparatus so a third open-circuit rig was constructed, Fig. 3. This consisted of a positive displacement air compressor, supplying air to the diode via a flow-measurement orifice. The compressor was capable of supplying air to the diode at a Reynolds No. of 1.5×10^5 (based on diode diameter), against a pressure drop of 20 lb/in^2. The majority of the new experimental work described here was undertaken using this equipment.

Flow visualisation was carried out by means of air bubbles in the previously described closed-circuit water apparatus. Flow directions could also be assessed in the open-circuit air rig due to dust particles passing through the apparatus becoming deposited on the walls of the diode.

4. Analysis of the performance of the fluid diode.

An attempt was made to express the performance of the diode in terms of ℓ/D and p/D. The approach to the problem consisted of a combination of analytical methods and experimental results. The detailed analysis is given in Appendix I.

The reverse-flow coefficient was calculated on the assumption that each cascade of blades performed as if it was placed within a two-dimensional flow. The alignment of the blades to the axis of the diode, together with the measured flow angles at inlet and outlet from the blade cascades, enabled resolved components of the blade lift and drag to be calculated along the diode axis. These forces, together with the additional drag force caused by the helical flow on the walls and centre bar of the diode, enabled a reverse-flow coefficient to be determined.

A second expression for the reverse-flow coefficient was obtained by considering the momentum changes of the fluid when passing through the cascades. Equating these two values of loss coefficient enables an equation to be obtained for the flow angles.

In the case of the forward-flow coefficient direct recourse to the experimental results had to be made. This was partly due to a lack of generalised experimental data being available for the blade geometry investigated. A further difficulty was that a combined analytical and empirical approach failed to predict sufficiently accurately the change of the forward-flow coefficient with change in helix angle.

5. Discussion of results

5. 1. Graphical presentation

The performance of the diode, Σ, is defined as the ratio of the resistance coefficient in the reverse-flow direction to the resistance coefficient in the forward-flow direction. Both resistance coefficients are determined when the flow is such that is has become independent of Reynolds number. The Reynolds number at which this occurs is dependent on the length of the diode, but has a value of approximately 10^5.

The resistance coefficient K is the ratio of the head loss measured across the diode to the mean velocity head in the pipe at entry to the diode.

$$K = \frac{2g\Delta P}{\rho U^2}$$

Dimensional analysis of the mode of operation of the diode indicates that

$$\frac{\Delta P}{\rho U^2} = F\left(\frac{\rho UD}{\mu}, \delta, \theta, n, \frac{\ell}{D}, \frac{p}{D}\right)$$

(see notation)

where $\frac{\rho UD}{\mu}$ is the Reynolds number,

and $\frac{\Delta P}{\rho U^2}$ is proportional to the Resistance Coefficient Factor.

The experimental investigation was carried out so that the effect on the resistance coefficient of changes in one dimensionless parameter was obtained while maintaining the remaining groups constant.

Figs. 4, 5 and 6 are representive graphs of the results showing K_F, K_R and

Σ plotted against Reynolds number. The conclusion to be drawn from these graphs is that K_F, K_R, and Σ become constant at a Reynolds number of approximately 10^5. The subsequent graphs show relationships which are only applicable to the Reynolds-number range above 10^5.

Fig. 7 shows the variation of K_F with ℓ/D. The majority of these results were obtained from the open-circuit apparatus, and for a single value of pitch, $p/D = 0.109$. The graph is linear, but does not pass through the origin. This type of characteristic is frequently obtained when a series of similar objects are placed behind each other, see Idel'chik (3). The shape of this graph is of primary importance in the determination of the form of the Σ versus ℓ/D function.

Also shown in Fig. 7. are values of K_F obtained with the blades inclined at angles of plus and minus 10° to a normal through the axis. Although only two points were obtained for each of these graphs it can be seen that in each case the gradient of the line is less steep than when the blades are positioned at right angles to the axis. The reasons for this are discussed later. Fig. 8 shows the variation of K_F with ℓ/D for different values of pitch and two helix angles. At present the most comprehensive series of results exists for $p/D = 0.109$ and $\theta = 90^{\circ}$. A reduction of the helix angle from 90° to 60° results in a reduction in K_F. Changes in pitch, which were mostly carried out in the previous work (1), do not change K_F markedly over the range tested.

Figure 9 shows the variation of K_F with ℓ/D for different numbers of cascades. The gradients of these lines are a non-linear function of the number of cascades since each additional cascade causes a reduction in flow area, and hence an increase in the velocity of flow.

The variation of the reverse-flow coefficient with ℓ/D, and p/D is shown in Fig. 10. For a given ℓ/D, K_R increases with decreasing p/D. This trend was observed previously (1) but is now shown over an extended range. Since the main experimentation has been with $p/D = 0.109$, results for this pitch cover a wider range than for the other values, and thus these results are shown separately in Fig. 11. For $\ell/D > 2$ there is clearly a linear relationship between K_R and ℓ/D for $p/D = 0.109$, and $\theta = 90^{\circ}$. Also shown is the relationship between K_R and ℓ/D for $\delta = +10^{\circ}$, and -10°. Although these lines are drawn through only two points each, and hence their linearity is not established it is clear that the value of K_R has been very substantially reduced. Also shown are a few results for different helix angles. It would appear that the effect on values of K_R of increasing the helix angle from 0° to 60° is not as great as a change from 60° to 90°.

The purpose of the helix is to aid the formation of the rotating flow within the diode, and hence the value of K_R. At the same time, however, K_F could be increased causing a reduction in performance below that which would otherwise be obtained. It was thought possible that once the helical flow has been obtained it would continue without the use of a helix. In order to test this idea an experiment was carried out in which the initial blades of the diode were placed on a helix, and the remainder were placed on a straight central bar. (see Fig. 10). Although the flow continued to spin there was a reduction in the value of the reverse-flow coefficient. A second experiment in which the number of blades on the straight bar was increased caused a further decrease in the value of the coefficient. Further analysis of Fig. 10 suggests that there were insufficient blades used on the helical central bar to develop fully the spinning flow before its transference to the straight section. Fig. 8 shows no significant change in the forward-flow loss coefficient.

The variation of K_R with ℓ/D for two and three cascades is compared with the results for four cascades in Fig. 12. The gradient of the linear section of the curve of the results for two cascades is approximately 1/8 the gradient of the curve for four cascades.

As both K_F and K_R vary linearly with ℓ/D it is a simple matter to calculate the performance as $\ell/D \to \infty$, assuming that these relationships do not change. The limiting value of the performance is 354 and is shown on Fig. 13. The curve on Fig. 13 shows the variation of performance with ℓ/D, and has been calculated from the K_F and K_R versus ℓ/D curves.

Previous work (1) has shown that for a given number of blades the optimum p/D ratio is 0.109. For a given length, however, the smaller the pitch the greater the number of blades, and since the forward-flow coefficient is little affected by pitch the overall result is to raise the performance. Fig. 14 shows Σ plotted against ℓ/D for various values of pitch and constant helix angle.

The shapes of the performance curves plotted against ℓ/D, Figs 15 and 16, with n, and θ as further parameters are dependent on the shapes of the K_R and K_F versus ℓ/D curves already discussed. In general Σ increases with ℓ/D with a further increase in performance brought about by raising the helix angle or the number of cascades. For a given length of diode the variation of performance with helix angle is clearly shown in Fig. 17.

The fluid leaves the diode with a spinning motion around the central axis. Therefore it must be questioned whether the use of static-pressure tappings in the wall of the diode can be used to determine its performance accurately? The effect of the rotation would be expected to cause a positive pressure gradient from the central axis to the outer wall. Thus the use of wall pressure tappings could be expected to underestimate the performance of the diode. In order to test this a pipe of length 75 pipe diameters was placed on the downstream side of the diode outlet diffuser, in order to allow the spinning motion to be attenuated by friction. Measurements of the pressure loss across the diffuser and outlet pipe indicated that at the maximum flow rate tested the reverse-flow coefficient was raised by 5%. It was considered that this was sufficiently small for no corrections to be made to the values of loss coefficient. In order to raise the performance of the diode by 5% it would be necessary to remove the swirl and add no further resistance in the forward-flow direction. An experiment in which a matrix type of flow straightener was fitted immediately after the diode showed no increase in the reverse-flow coefficient, although the swirl was considerably reduced.

5.2. Theoretical predictions

Appendix I estimates that the value of the loss coefficient due to the helical flow on the diode wall and central bar is $0.38\,(\ell/D)^{0.8}$ This value is applicable for a flow Reynolds number of 10^5 and zero stagger angle. The stagger angle is the inclination of the blades to a normal passing through the cascade axis. For a circular pipe with a relatively smooth surface the loss coefficient could be expected to be $0.035\,(\ell/D)$, again for a Reynolds number of 10^5. Hence the calculations estimate that the helical flow path has caused the friction loss to be increased by a factor of approximately ten. Experimentally it was not possible to separate the wall shear losses from those produced by the blades, as the blades must be in position to impart the rotating flow. Measurement of the total losses, however, indicates that the friction losses are small compared with those produced by the blades.

For the case of zero stagger angle the gradient of the K_R versus ℓ/D graph, Fig. 11, was found to be 510 in the linear region. This can be compared with 538 and 400, predicted by the blade theory and the momentum method respectively. It should be noted that the measured flow angles, β_1, and β_2, employed in the calculations, only apply to the linear region. For a stagger angle of $+10^{\circ}$ the two points on Fig. 11 give a gradient of 160. For this stagger angle the blade theory and momentum methods predict values of gradient of 357, and 230 respectively. It is possible, that for $\delta = +10^{\circ}$, the value of ℓ/D at which the gradient of the K_R versus ℓ/D graph becomes constant may be greater than for when $\delta = 0$. Thus the two points on Fig. 11 may not indicate the gradient of the linear part of the curve. Therefore, a direct comparison of the measured and predicted gradients for $\delta = +10^{\circ}$ may not be valid. It can be noted, however, that the change in stagger angle has resulted in a reduction of the gradient, and that the measured change in flow angles results in lower gradients being predicted by equations (5) and (14).

Although, for zero stagger angle, there is good agreement between the measured value of the gradient and that predicted by the blade theory, this is not so for the prediction by the momentum method. It should be remembered that assumptions of two-dimensional flow, and constancy of flow angles along the span of the blades, were made. Analysis of the equations shows that the value of the gradient is sensitive to the flow angles, thus greater accuracy could be obtained by measuring the mean angle of flow rather than the direction of the wake behind the aerofoils. Further inaccuracies have been introduced by the use of a mean velocity of flow. The actual flow paths are such that the effective mean velocity may differ from that given by equation (2).

Solution of equation (15) for zero stagger angle gives a value of 37° for the flow inlet angle. This can be compared with the value of 30° determined experimentally.

For the experiments in which the stagger angle was -10° it was observed that considerable separation of the flow was taking place from the convex surface of the aerofoils, and that the separation was initiated close to the leading edge. A consideration of the geometry of a diode with cascades whose stagger angle is negative shows that the fluid is flowing to a region of higher pressure than that which is prevailing on the inlet side of the cascade. This adverse pressure gradient is the cause of the early separation, and renders the analysis invalid for negative stagger angles. The flow separation reduces the volumetric flow rate over the aerofoils by means of a reduction in the effective area between pairs of blades. This, together with any adverse effect on the lift coefficient, results in reduced axial forces being developed, and hence reduces the reverse-flow coefficient. The reduction in reverse-flow coefficient can be seen in Fig. 11.

6. Conclusions

The performance increases with the length of the diode. The maximum value determined experimentally was 181 for $\ell/D = 6$. The linear variation of K_F and K_R with ℓ/D suggests that an asymptotic value of the performance of 354 should be approached for large values of ℓ/D. The above figures are applicable for a $p/D = 0.109$, 4 cascades, and Reynolds number in excess of 10^5.

The best performance is obtained with the blades set at right angles to the diode axis, and for a given number of blades the use of four cascades is better than three or two.

It is possible to predict theoretically the approximate rate of change of reverse-flow loss coefficient with ℓ/D, in the region in which the flow angles remain constant. The analysis is also sufficiently representative of the flow to demonstrate the change of loss coefficient brought about by changing the blade stagger angle.

7. Acknowledgements

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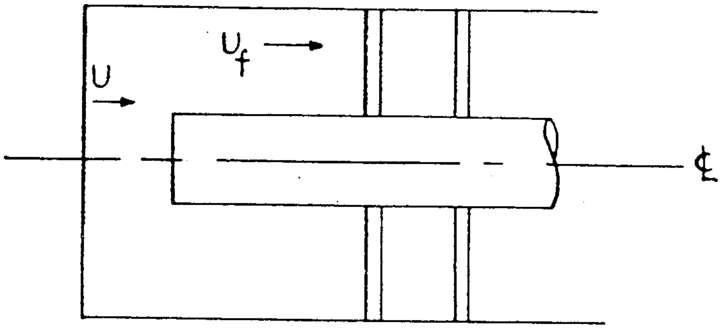
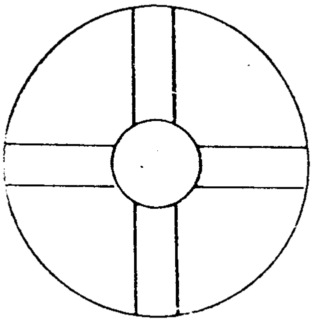
Appendix

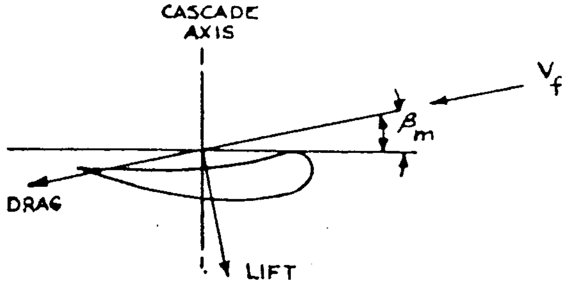
Reverse-flow loss coefficient

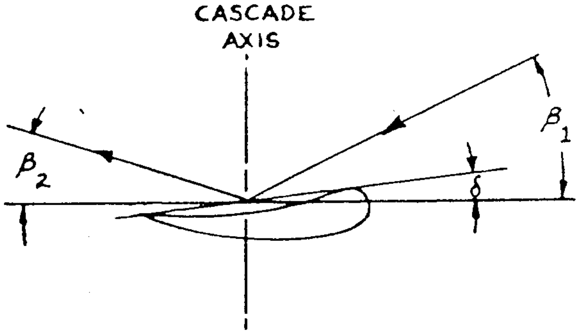
a) Pressure loss due to aerofoils





Let F be the total force/blade in the direction of the centre line due to aerodynamic lift and drag.





$$F = \rho \frac{V_f^2}{2g} a \left[C_L \cos\beta_m + C_D \sin\beta_m \right] \qquad \text{--------------(1)}$$

It is assumed that the flow over the cascades is two dimensional, and that the mean flow angle is applicable over the full span of each blade, and over the length of the diode under consideration.

Where V_f = velocity of fluid approaching blades.
a = Area of each blade.
C_L = lift coefficient of blade.
C_D = drag coefficient of blade.
β_m = mean flow angle, determined from blade inlet and outlet angles β_1 and β_2 respectively.
ρ = density of fluid.

tan β_m is given by

$$\tan\beta_m = \tfrac{1}{2}\left[\tan\beta_1 - \tan\beta_2\right] \text{ Ref. (4).}$$

V_f can be related to the mean velocity of flow, U, in the inlet section of the diode by the equation of continuity.

$$UA = V_f A_f \sin \beta_m \qquad \text{-------------------------------}(2)$$

where A represents the cross-sectional area of the inlet section of the diode, and A_f represents the free flow area between the blades.

$$A_f = \frac{\pi}{4}(D^2 - d^2) - \frac{n}{2}(D-d)c$$

where D = diameter of diode
d = diameter of central bar
c = blade chord
n = number of cascades

For the size of diode tested in these experiments

$$\frac{A}{A_f} = 1.7, \text{ and } \therefore \; V_f = \frac{1.7U}{\sin\beta_m}$$

The loss coefficient is defined as the ratio of the head loss across the diode to the upstream velocity head.

$$\text{Thus } K_R = \frac{2\,gn\,N\,F}{\rho U^2 A_f} \qquad \text{-------------------------------}(3)$$

where N = the number of blades/cascade
and g = the acceleration due to gravity.

Substituting equation (1) and (2) into (3), and re-writing A, and A_f in terms of D, d, and c, the following is obtained.

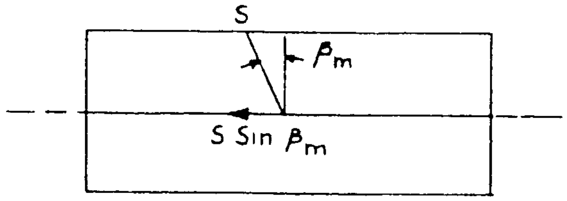
$$K_R = n\,N\left(\frac{A}{A_f}\right)^2 \frac{c(D-d)}{2A_f} \operatorname{Cosec}\beta_m \left[C_L \operatorname{Cot}\beta_m + C_D\right] \text{-----------}(4)$$

or, in terms of the length and pitch of the diode:

$$K_R = n\left(\frac{A}{A_f}\right)^2 \frac{c(D-d)}{2\,A_f}\,\frac{D}{P} \operatorname{Cosec}\beta_m \left[C_L \operatorname{Cot}\beta_m + C_D\right]\frac{\ell}{D} \text{ ------ }(5)$$

The pressure losses associated with the sudden contraction and expansion of the flow, at inlet and outlet from the diode, have been ignored as they are small compared with those produced by the aerofoils.

b) Pressure losses due to shear stresses at diode wall and central bar.



Development of wall of diode.

The flow within the diode is helical, and thus the resolved part of the shear force S must be obtained to determine the pressure drop due to friction.

Suppose the wall shear stress is τ, then:

$$S = \tau \pi D \ell \quad \text{for the outer wall}$$

Similarly the central bar also contributes to the friction an amount $\tau \pi d \ell$

Thus the total wall shear stress is given by

$$S_t = \tau \pi \ell (D + d) \qquad \text{------------------------(6)}$$

Equating the pressure forces across the diode to the shear forces gives

$$\Delta P \frac{\pi}{4}(D^2 - d^2) = \tau \pi \ell (D + d) \sin \beta_m$$

or

$$\frac{\Delta P}{\ell} = \frac{4}{(D-d)} \tau \sin \beta_m \qquad \text{----------------------(7)}$$

The shear stress can be written in terms of a friction factor

$$\tau = C_f \frac{\rho V^2}{2g} \qquad \text{------------------------(8)}$$

If we consider the flow over the walls of the diode to be similar to that over a flat plate it then becomes possible toestimate the shear stress.

For the turbulent flow over a flat plate

$$C_f = 0.074 \; Re_f^{-0.2} \qquad \text{------------------------(9)}$$

Where Re_f = the Reynolds number based on the length of the flow path.

Substituting (8) and (9) into (7) gives

$$\Delta P = \frac{4\ell}{(D-d)} \rho \frac{V^2}{2g} \times 0.074 \; Re_f^{-0.2} \; \sin \beta_m$$

Re-writing V in terms of U, and Re_f in terms of Re gives:

$$K_{SF} = \frac{2g\Delta P}{\rho U^2} = \frac{4D}{(D-d)} \times \frac{0.074 \times 1.7^{1.8}}{(\sin \beta_m)^{0.6}} \left(\frac{\ell}{D}\right)^{0.8} Re^{-0.2} \qquad \text{----(10)}$$

The total reverse flow loss coefficient is given by the sum of equations (5) and (10).

c) Predicted numerical values of K_R

The experimental diode had the following dimensions:

$\frac{A}{A_f}$	1.7
c	1 in.
D	5.5 in.
d	1.5 in.
n	4
p	0.6 in.

The inlet and outlet flow angles, β_1 and β_2 respectively, were measured, and are shown in the following table.

Also shown are the values of β_m, C_D, C_L, and the calculated value of $K_R \frac{D}{\ell}$ for two values of stagger angle.

K_R due to aerofoil lift and drag.

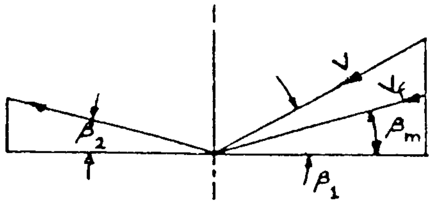
δ^o	β_1^o	β_2^o	β_m^o	C_L	C_D	$K_R \frac{D}{\ell}$
0	30	10	11.3	1.38	.067	538
10	22.5	0	11.7	.98	.064	357

Taking a typical Reynolds number of 10^5, the loss coefficient due to shear forces from equation (10) is·

$$K_{SF} \quad = \quad 0.38 \left(\frac{\ell}{D}\right)^{0.8} \text{ for } \beta_m = 11.3^o$$

Thus the contribution towards the loss coefficient is very small, and in many cases it can be neglected.

d) The determination of K_R from momentum changes.



Flow between a pair of blades

The change in the axial velocity of the fluid is given by:

$$\Delta U_f \quad = \quad V_f \operatorname{Cos}\beta_m (\tan\beta_1 + \tan\beta_2) \qquad \text{---------------(11)}$$

The geometry of the blades is such that the inlet velocity can be taken as V,

$$\text{where} \quad V \quad = \quad V_f \frac{\operatorname{Cos}\beta_m}{\operatorname{Cos}\beta_1} \qquad \text{---------------(12)}$$

Therefore, the rate of change of momentum in the axial direction caused by fluid passing between a pair of blades

$$= \frac{\rho V_f^2}{2g} (D-d)p \frac{\operatorname{Cos}^2\beta_m}{\operatorname{Cos}\beta_1} (\tan\beta_1 + \tan\beta_2) \qquad \text{---------------(13)}$$

= the force F exerted on each blade.

Since there are nN blades, and the effective cross-secional area is A_f, the loss coefficient can be obtained by substitution into equation (3)

$$K_R \quad = \quad n\left(\frac{A}{A_f}\right)^2 \frac{D(D-d)}{A_f} \frac{\tan\beta_1 + \tan\beta_2}{\operatorname{Cos}\beta_1 \tan^2\beta_m} \frac{\ell}{D} \qquad \text{--------------- (14)}$$

Substitution of the measured values into equation (14) shows that

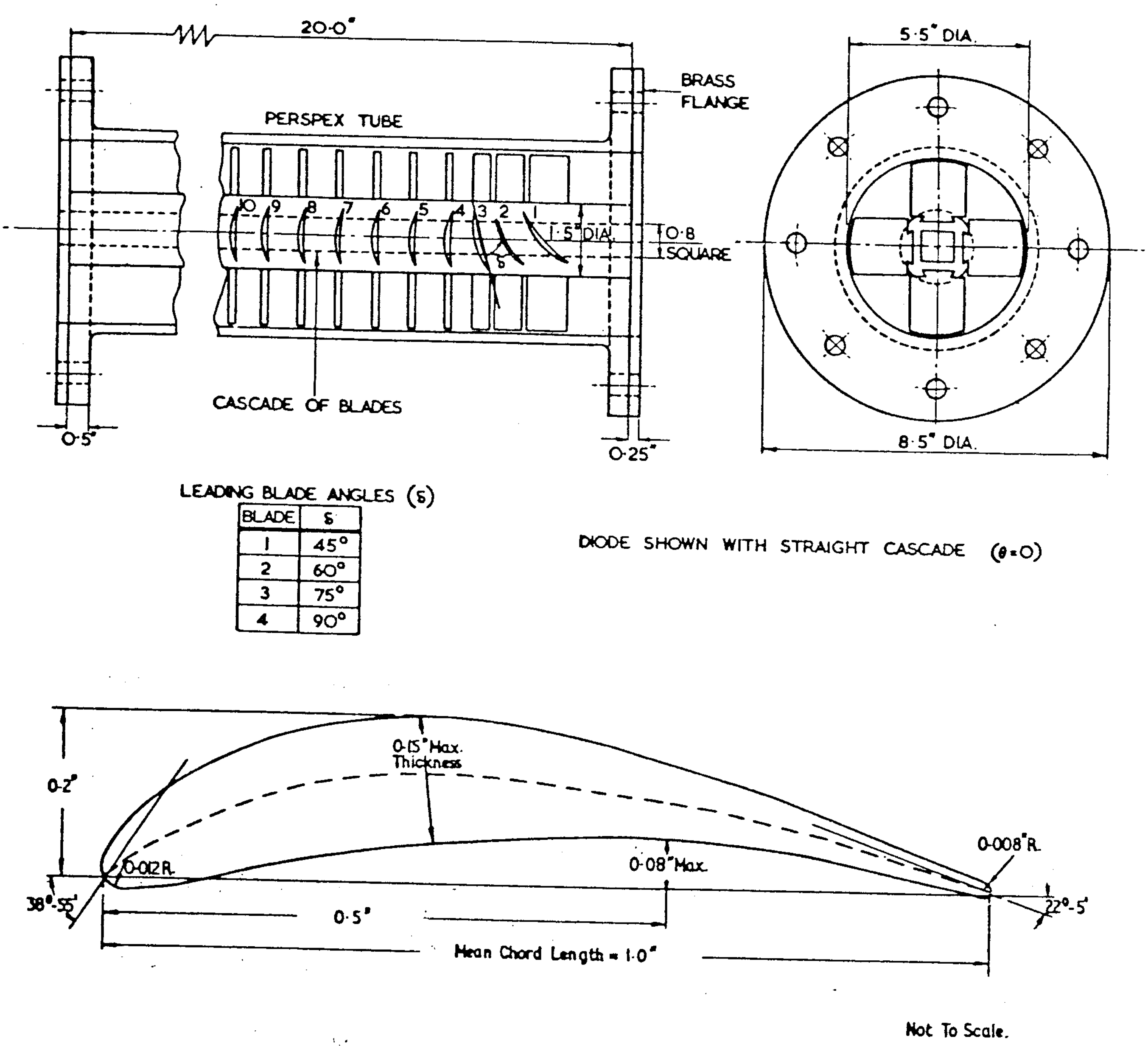
$$K_R \quad = \quad 400 \frac{\ell}{D}$$

e) Theoretical value of β_1

Equations (5) and (14) are two separate expressions for K_R. Elimination of K_R from these two equations, and assuming C_D is negligible compared with $C_L \operatorname{Cot} \beta_m$, results in:-

$$c\, C_L \operatorname{Cos} \beta_1 = 2p \operatorname{Cos} \beta_m (\tan \beta_1 + \tan \beta_2) \qquad \text{--------------(15)}$$

Solution of equation (15) shows $\beta_1 = 37^{\circ}$.



LEADING BLADE ANGLES (δ)

BLADE	δ
1	45°
2	60°
3	75°
4	90°

FIG. 1. FLUID DIODE AND BLADE PROFILE

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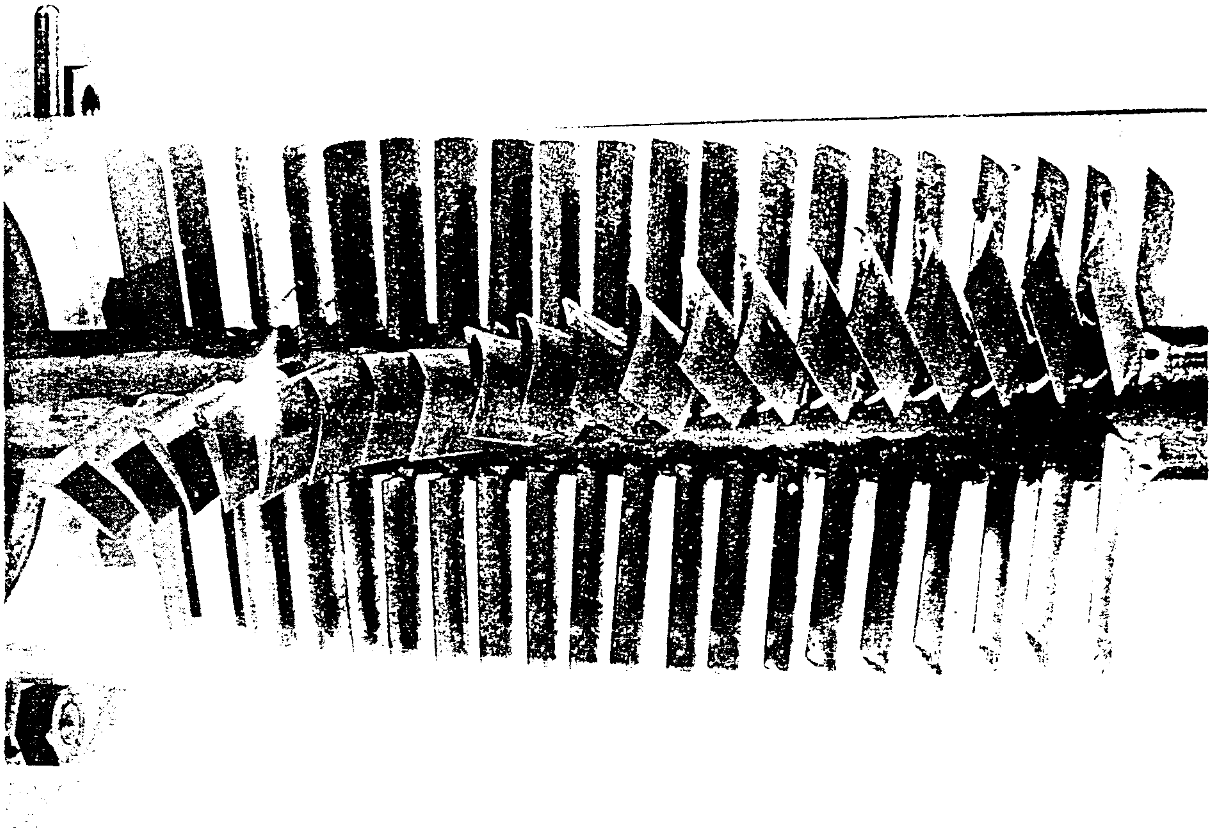
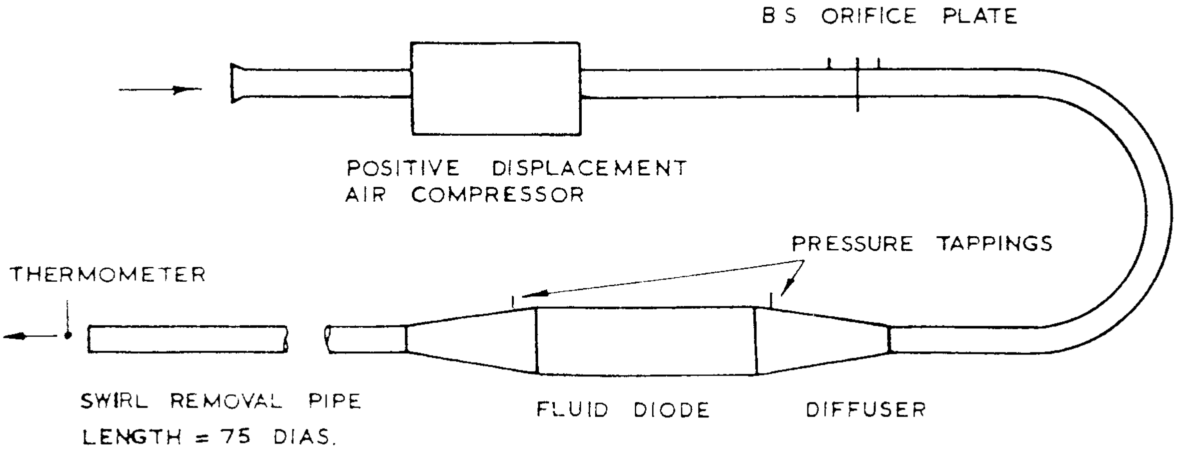


FIG. 2. FLUID DIODE



B S ORIFICE PLATE
POSITIVE DISPLACEMENT AIR COMPRESSOR
PRESSURE TAPPINGS
THERMOMETER
SWIRL REMOVAL PIPE LENGTH = 75 DIAS.
FLUID DIODE
DIFFUSER

FIG. 3. OPEN CIRCUIT TEST APPARATUS

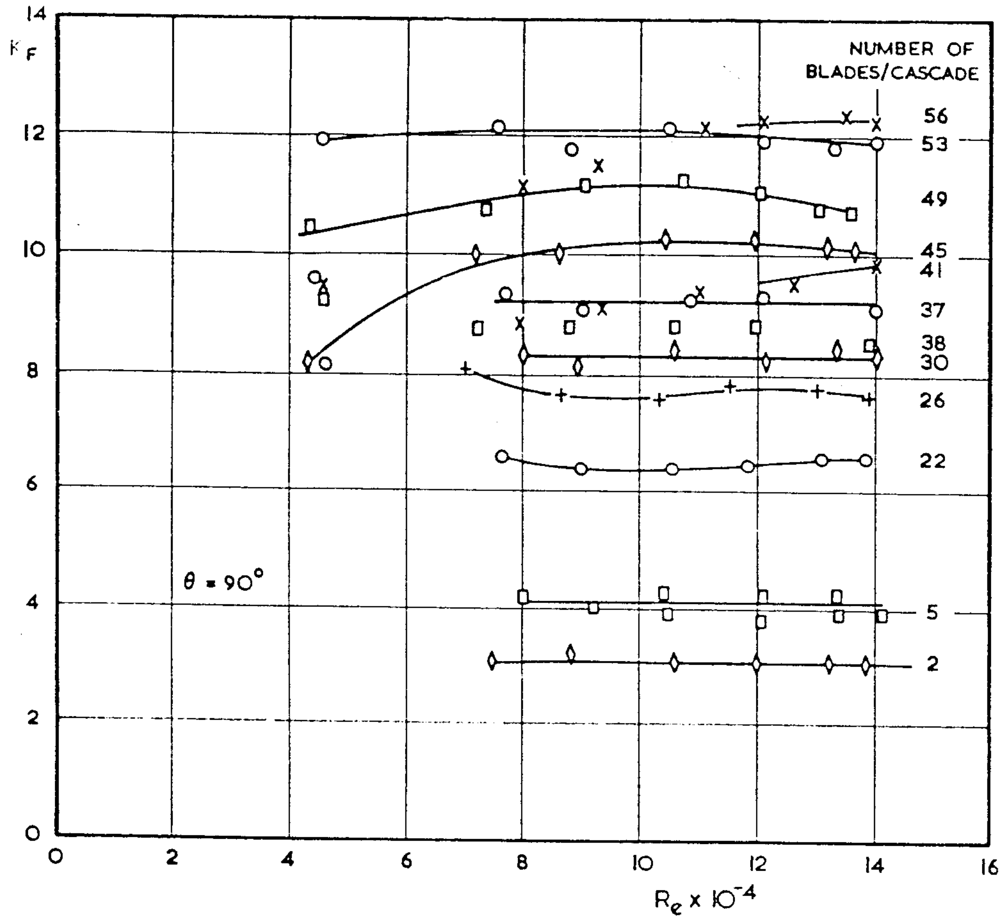


FIG. 4. K_F AS A FUNCTION OF REYNOLDS NUMBER, AND NUMBER OF BLADES

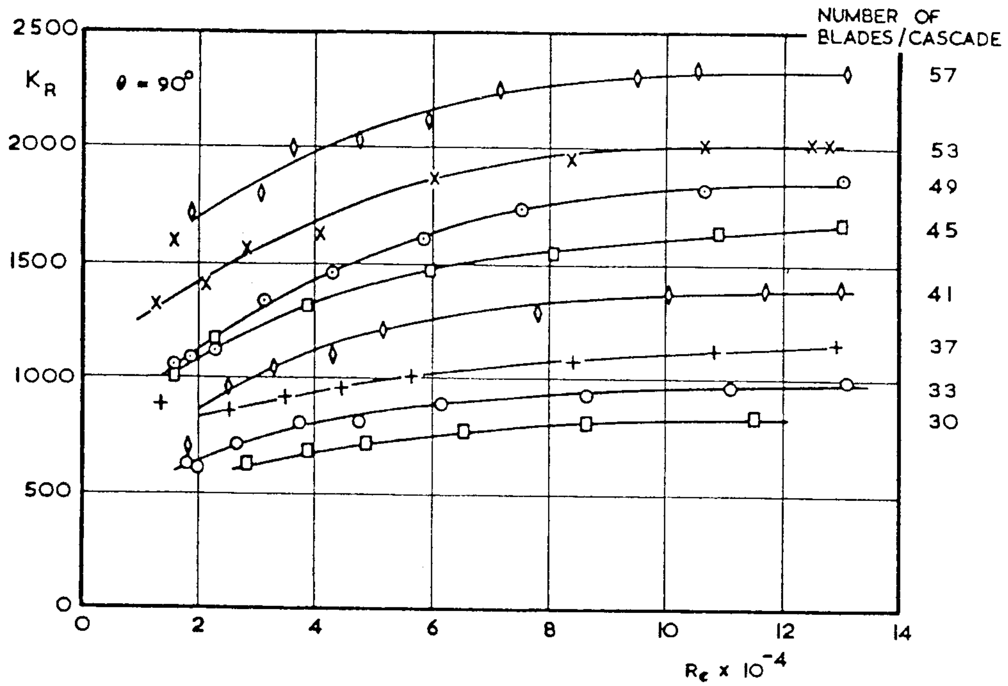


FIG. 5. K_R AS A FUNCTION OF REYNOLDS NUMBER, AND NUMBER OF BLADES

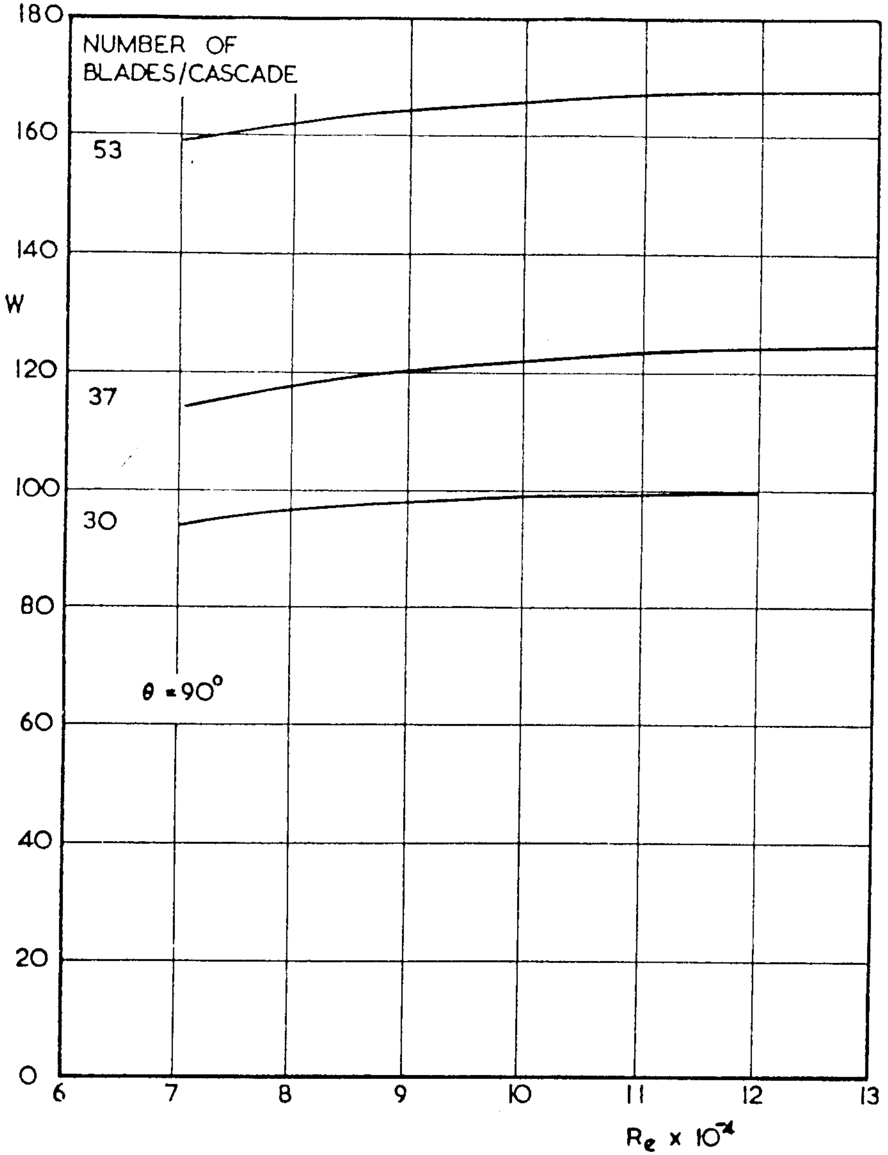


FIG. 6. Σ AS A FUNCTION OF REYNOLDS NUMBER, AND NUMBER OF BLADES

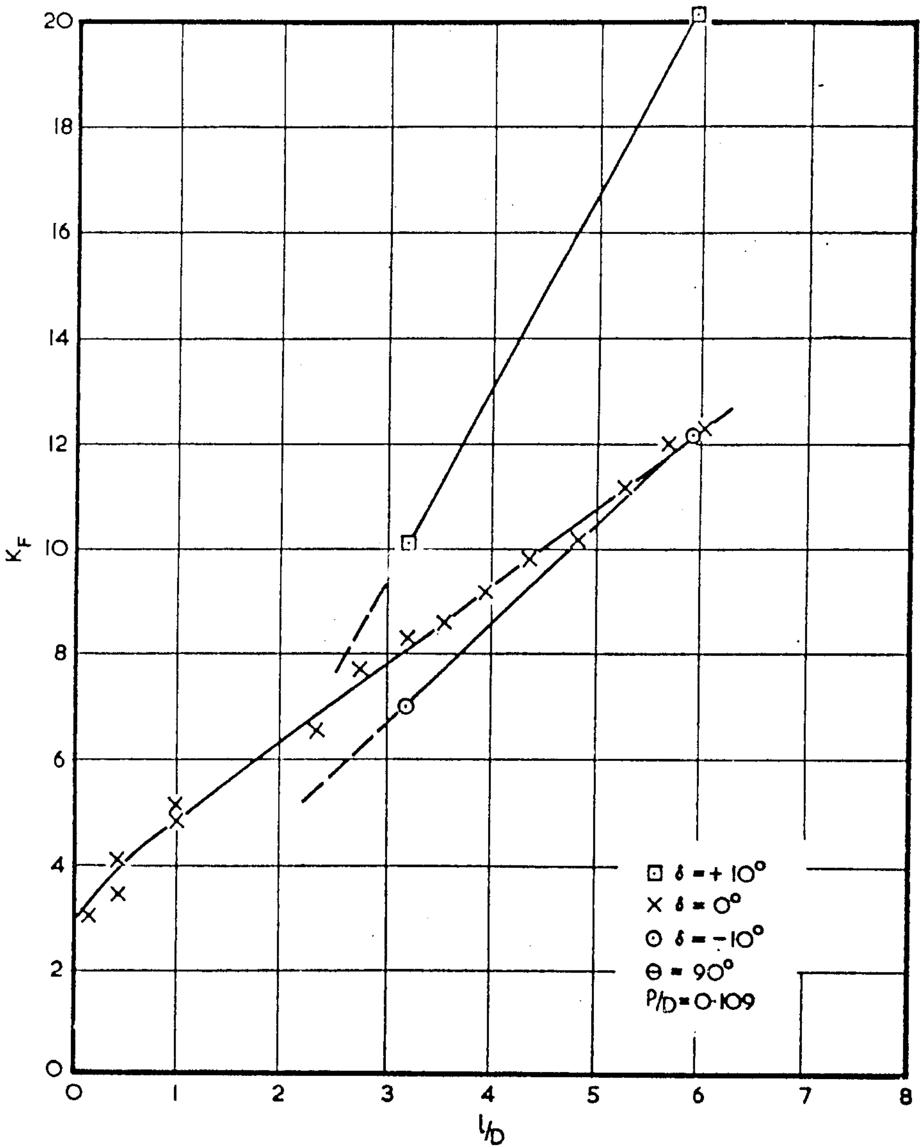


FIG. 7. THE VARIATION OF K_F WITH ℓ/D

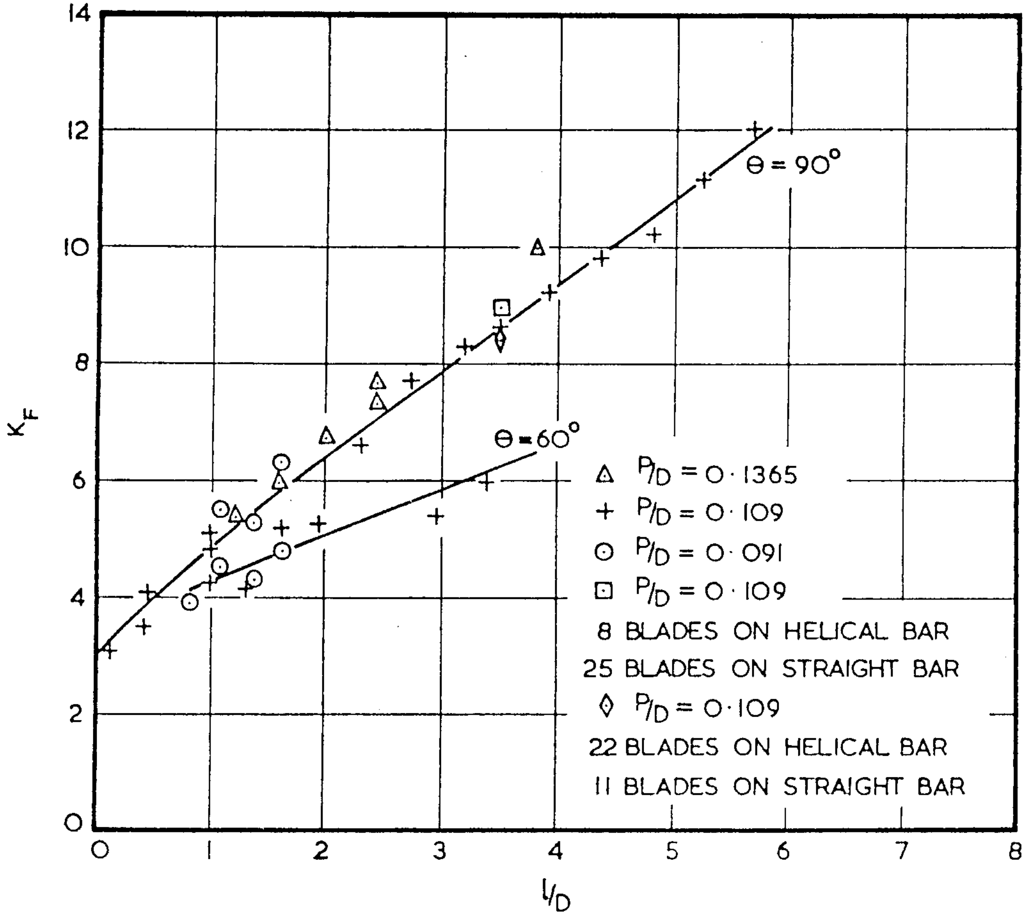


FIG. 8 THE VARIATION OF K_F WITH ℓ/D, θ, AND p/D

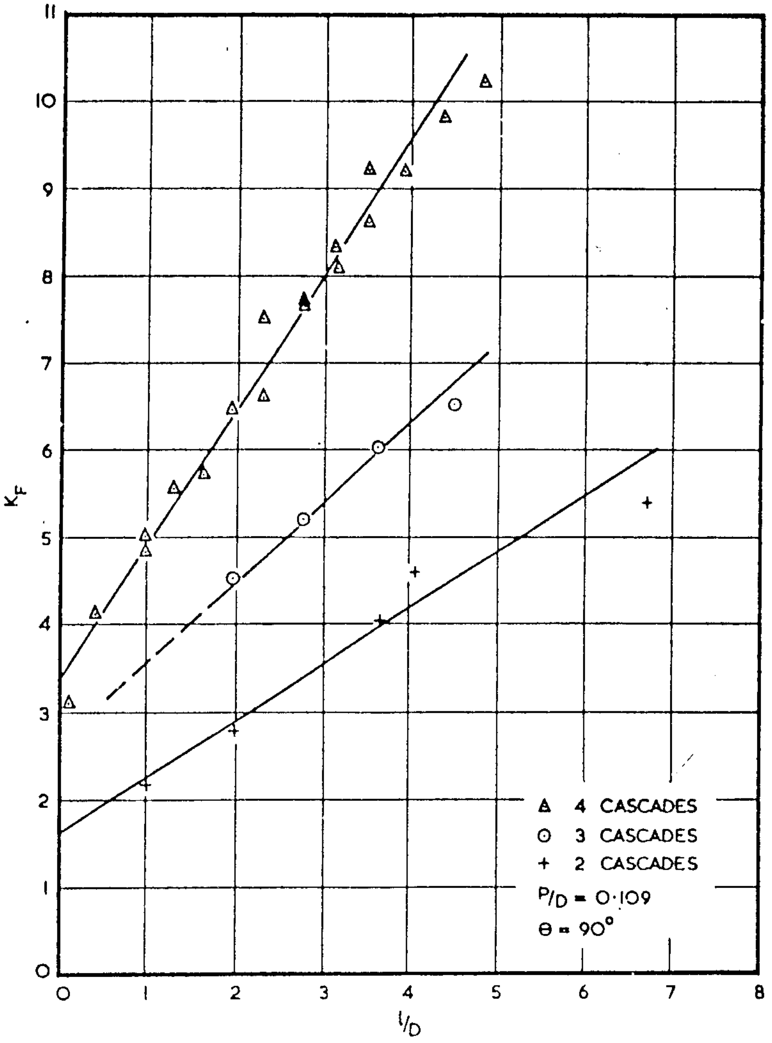


FIG. 9 THE VARIATION OF K_F WITH ℓ/D AND n

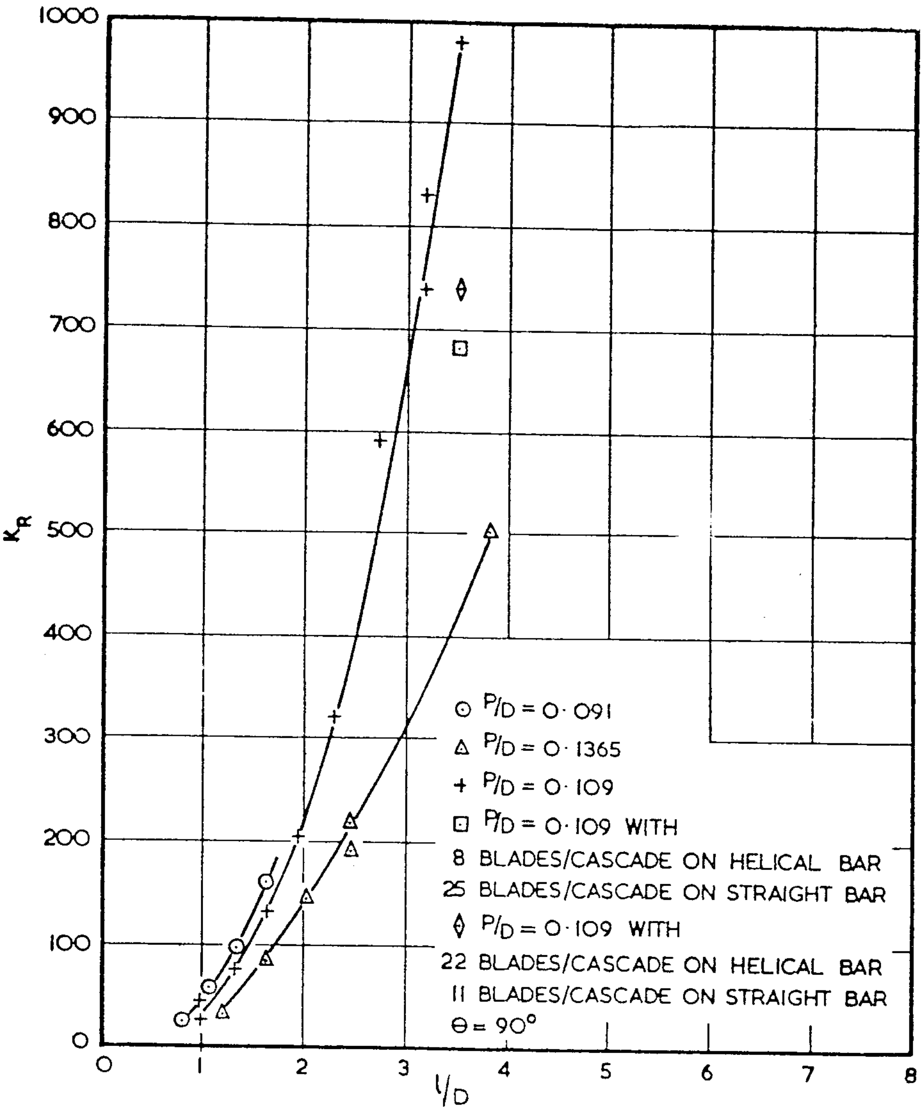


FIG. 10 THE VARIATION OF K_R WITH ℓ/D AND p/D

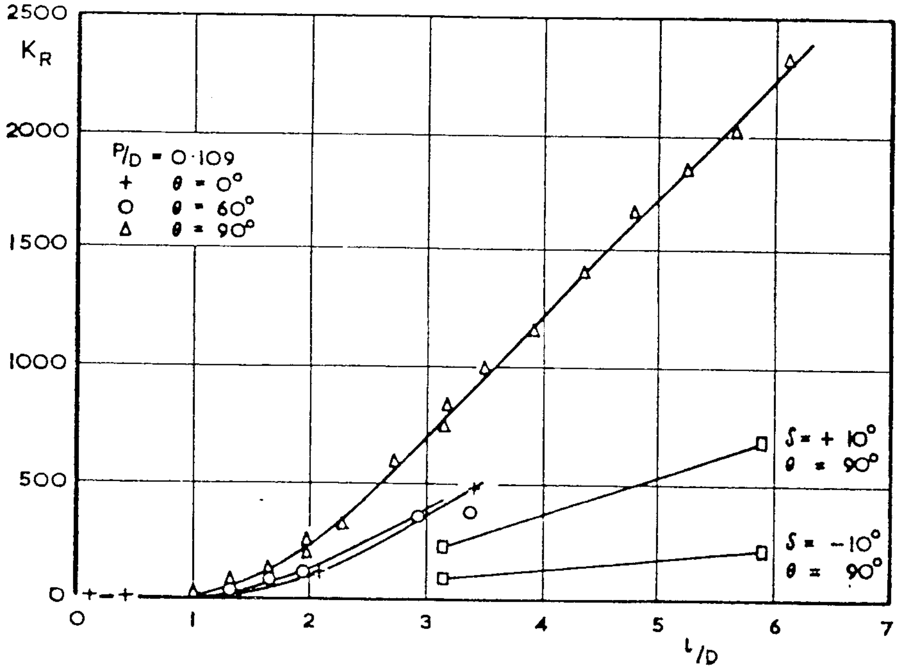


FIG. 11 THE VARIATION OF K_R WITH ℓ/D AND θ

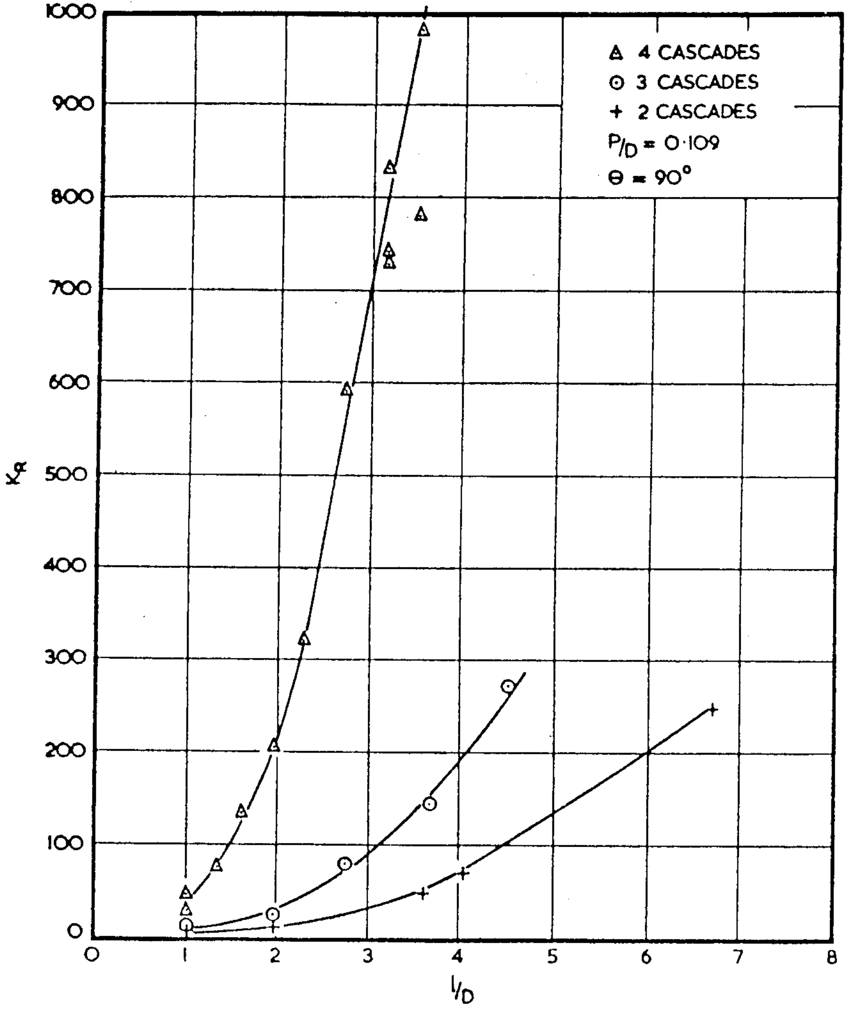


FIG. 12 THE VARIATION OF K_R WITH ℓ/D AND n

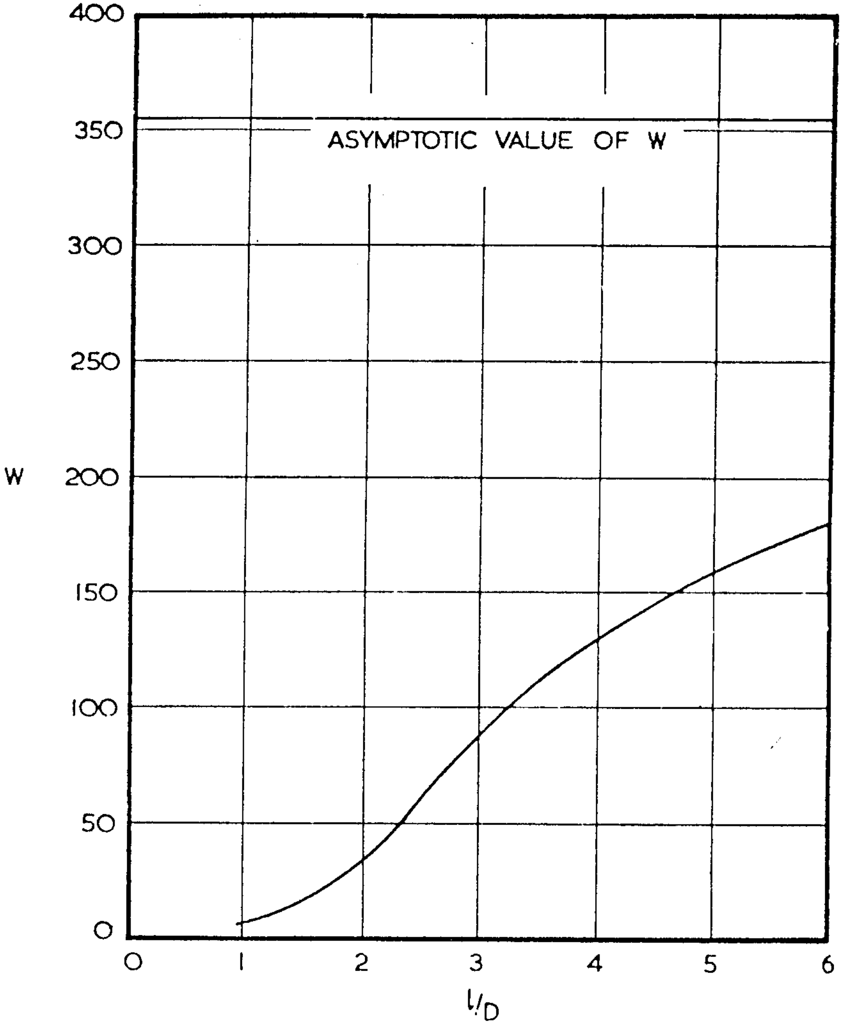


FIG. 13 THE VARIATION OF Σ WITH ℓ/D

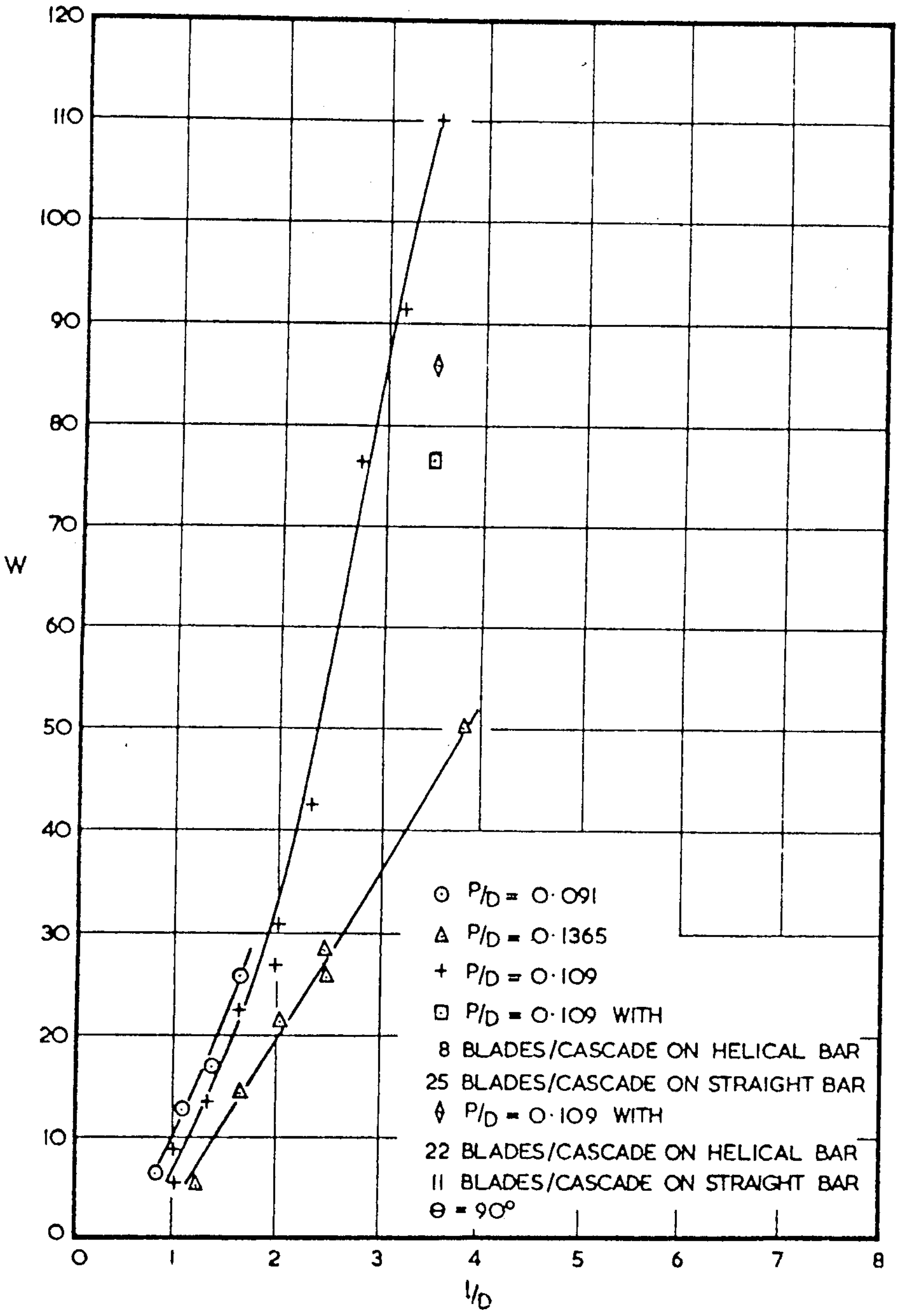


FIG. 14 THE VARIATION OF Σ WITH ℓ/D AND p/D

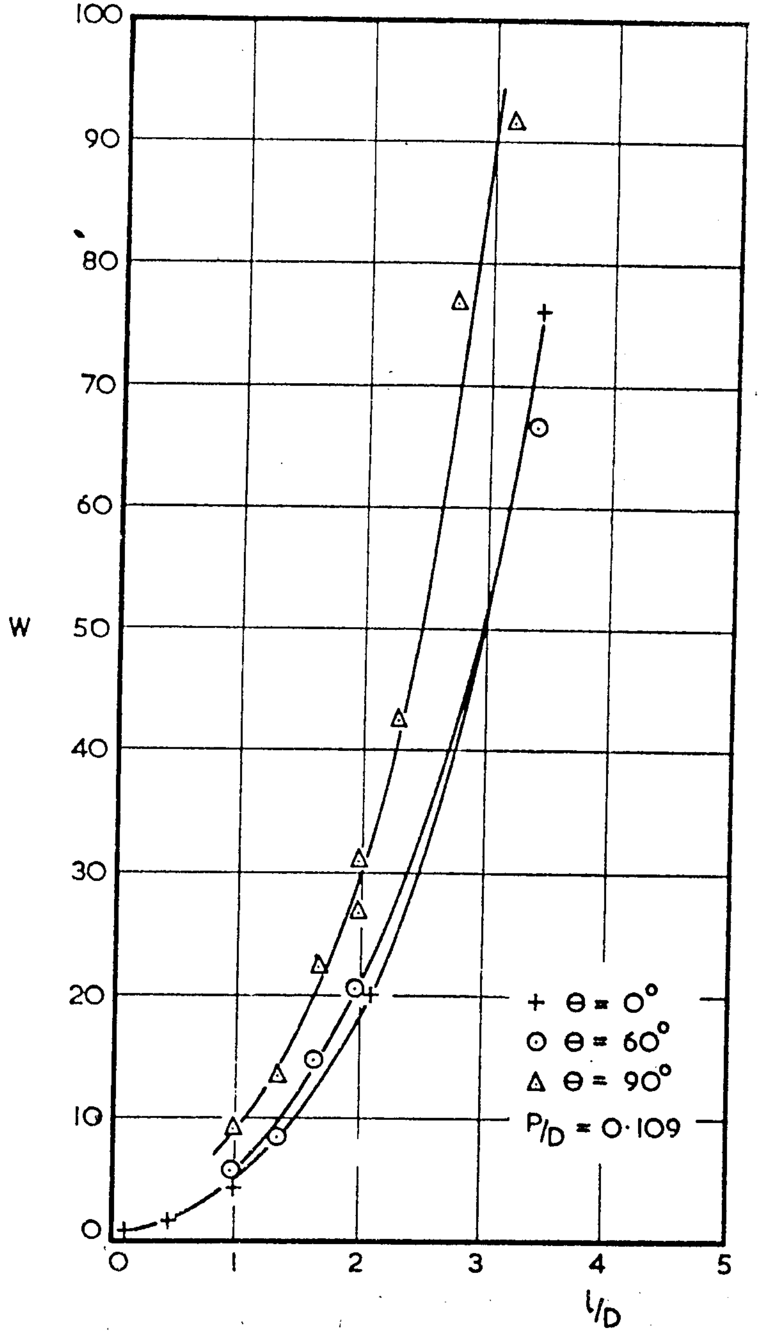


FIG. 15 THE VARIATION OF Σ WITH ℓ/D AND θ

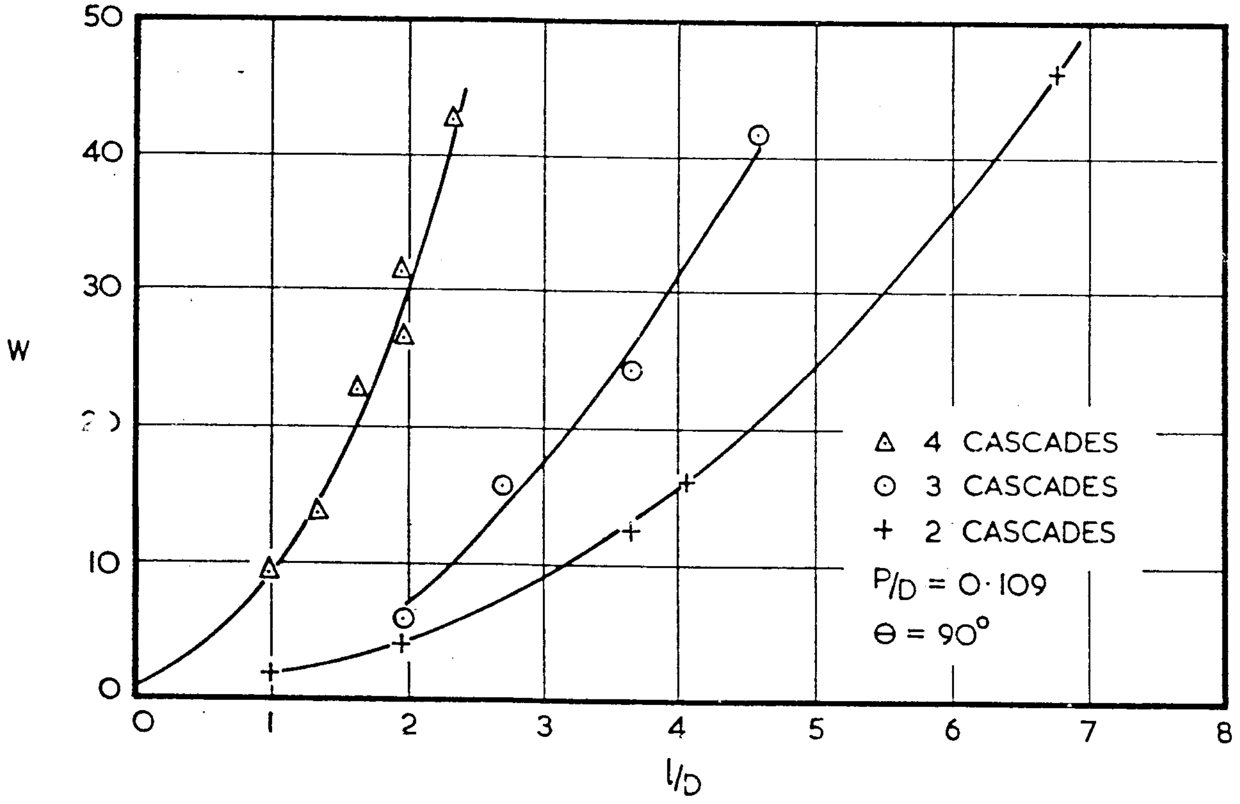


FIG. 16 THE VARIATION OF Σ WITH ℓ/D AND p/D

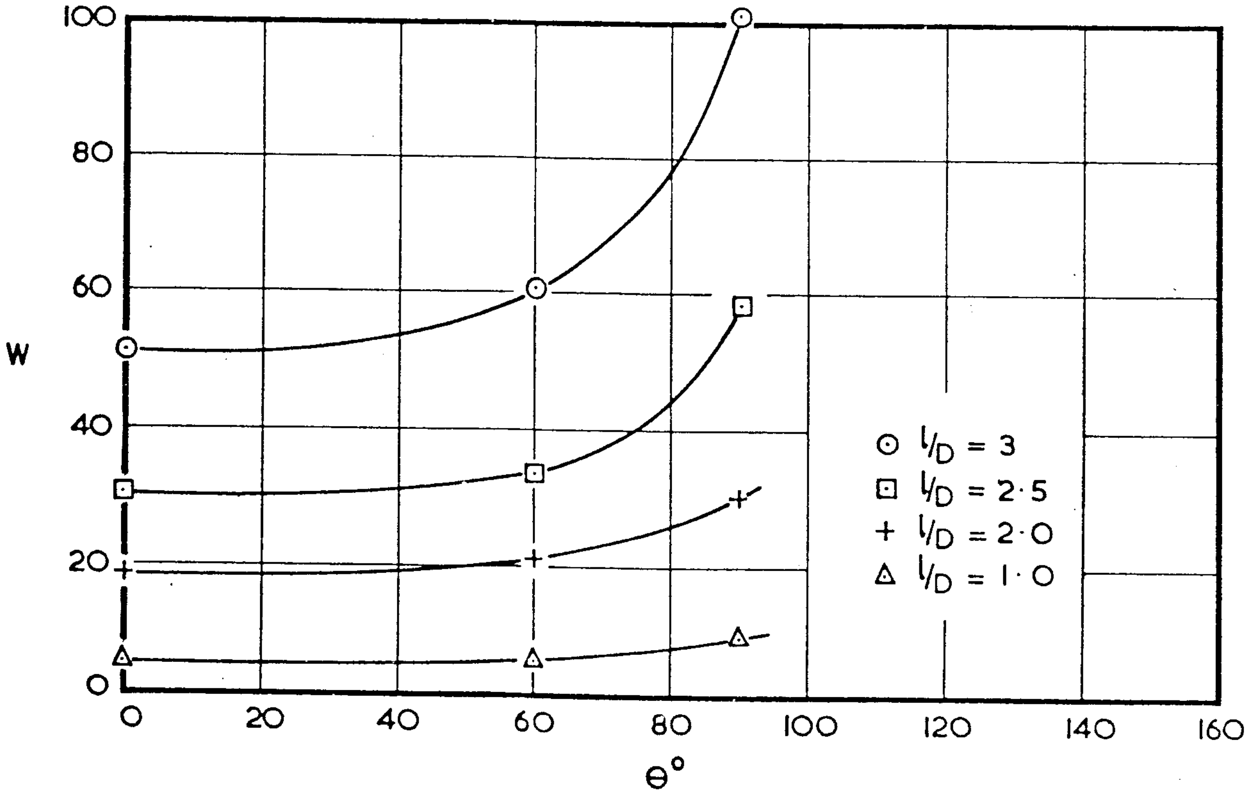


FIG. 17 THE VARIATION OF Σ WITH θ FOR GIVEN VALUES OF ℓ/D

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SECOND CRANFIELD FLUIDICS CONFERENCE
3rd – 5th January, 1967. CAMBRIDGE

PAPER D6

p. D6-88 to D6-115

A COMPARISON OF FLUID DIODES

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ABSTRACT

A comparative study of three types of fluid diodes operating on entirely different principles is presented in this paper.

Particular emphasis has been placed upon correlating the performance defined as the ratio of the pressure drop in reverse-flow direction to the pressure drop in the forward-flow direction under steady-state conditions over a range of Reynolds numbers.

The results of geometrical modifications made to these three diodes in order to improve their performance are also reported.

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CORRECTIONS

Paper D6

p. D6-95 The first paragraph should be deleted.

p. D6-96 In CONCLUSIONS para. (c)
of :1 and, should be: of 31:1 and

p. D6-115 FLUID RECTIFIER after FIG. 36 should be deleted.

ACKNOWLEDGEMENTS

This work was initially undertaken, under contract, for the United Kingdom Atomic Energy Authority.

The continuation of this work was then partly financed by a U.K.A.E.A. Special Grant.

The help given throughout this work by Messrs.A.F. Taylor and P.T. George of the Atomic Energy Authority's Reactor Group at Risley is gratefully acknowledged.

NOMENCLATURE

A	Flow area
d	Pipe diameter
g	Gravitational constant
h	Step height (Scroll Diode)
L	Length of cascade
P	Blade pitch
p	Absolute pressure
Δp	Pressure drop across diode
$\overline{V}$	Mean fluid velocity at entry to diode
x	Width of annular gap between cup and venturi (scroll diode)
β	Leading blade angles
θ	Helix angle measured over 15" length
μ	Absolute viscosity
ρ	Density

Subscripts

F	Forward flow direction
R	Reverse flow direction
S	Static condition
i	At inlet section (Scroll diode)
t	At throat section (Scroll diode)
x	Flow section between cup and venturi (Scroll diode) at a 2" radius from axis

Dimensionless Parameters

$\Sigma = \frac{K_R}{K_F}$	Performance ratio.
$K = \frac{2g\Delta p}{\rho \overline{v}^2}$	Resistance coefficient factor.
$Re = \frac{\rho \overline{v} d}{\mu}$	Reynolds Number based upon inlet pipe diameter.

INTRODUCTION

Interest has recently been shown in the possibilities of using a fluidic device whose characteristics are that it imposes (a) low resistance to fluid flowing through it in one direction and (b) high resistance when the flow direction is reversed.

The main advantage of a fluid valve, or diode, over the usual mechanical type is that it has no moving parts which could fail completely at the instant of operation. Its main disadvantage is that it will not fully prevent the reverse flow.

Many fluid diode designs have been suggested, but a review of the literature shows that most work has been concentrated on the Vortex diode.

Heim (1) and Zobel (2) have carried out experimental investigations on the vortex diode in order to find its optimum performance. The result of Zobel's studies is that the ratio of reverse-flow pressure loss to forward-flow pressure loss can be as much as 50:1.

In order to assess the relative merits of the vortex diode and two other fluid diode designs, which work on different principles, an investigation into the performance of these latter designs was carried out. A limited number of tests were also made on a vortex diode.

The two other designs, namely the scroll diode and the fluid-flow rectifier, were based on patents by Linderoth (3) and Kantrowitz (4) respectively. Neither of these diodes had apparently been tested, although the performance figures claimed suggested that they would give a better performance than the vortex diode.

The work was initially started through a U.K.A.E.A. contract which predetermined the overall dimensions of the three diodes.

OPERATING PRINCIPLES OF THE FLUID DIODES

Vortex Diode (Fig.1)

In the high-resistance direction of flow the fluid enters the diode through the tangential port and forms a vortex. Owing to this vortex the flow achieves a high dynamic head at the centre of the chamber. This head is destroyed when the fluid discharges through the axial port in the centre of the chamber, this giving the high flow resistance.

In the low-resistance, or forward-flow direction, the fluid enters the chamber through the axial port and leaves through the tangential port without forming a vortex and only a small pressure loss.

Scroll Diode (Fig.2)

This diode works on the principle that in the high-resistance direction of flow, the high velocity jet at the throat of the venturi tube separates from the walls and enters the annular cup. The fluid then spins back on itself and is subjected to a further deflection through an angle of about 180° before it emerges from the diode through the restricted annular space between the cup and the venturi tube. In addition, the rapid expansion of the fluid behind the cup provides a further head loss.

In the forward or low-loss direction of flow, the fluid passes smoothly through the diode without entering the cup. The venturi tube should also be designed for maximum pressure recovery so that the overall resistance to flow through this device is a minimum.

Fluid Flow Rectifier (Fig.3)

In the forward-flow direction the space between the cascades provides an unobstructed passage through which the fluid passes. The blades at the entrance to the diode help to guide the flow into each of the passages. The small head loss in this flow direction is due mainly to the eddies formed on the downstream side of the aerofoil sections.

When the fluid flows in the reverse, or high-loss direction, the blades impart a circumferential direction of movement to the fluid. To initiate this helical flow, the first few blades are set at progressively greater angles to the axis of the rectifier. It is also preferable if the outlet angle of the majority of the blades is set to direct the fluid leaving the blades against the flow between the blade cascades.

A modified form of this design is to set the cascade on a helix. This gives the open passages a slight helix so that a component of circumferential flow is imparted to the forward flow stream through the rectifier.

In the reverse-flow direction the leading edge of each blade is unobstructed by the blade above. This permits the blade to impart circumferential flow to the fluid much more quickly than with the blades located directly behind one another.

The result expected is for a slight increase in resistance in the forward-flow direction and a much larger increase in the reverse-flow direction giving an overall improvement in performance.

EXPERIMENTAL APPARATUS AND PROCEDURE

During the early stages of this investigation a closed circuit water rig Fig.4 was used to test the diodes. The main advantage of using water as the working fluid was that visual observations of flow patterns could be made, either by dye injection through the walls of the diode or by observing the movements of small air bubbles in the water.

The centrifugal pump in this rig could produce a maximum head of 75 psia and steady flow rates up to 300 g.p.m. through the working section.

A small "Airhydropump" was also incorporated to raise the static pressure of the circuit to any desired level up to 60 p.s.i.

The majority of the results were obtained on a closed circuit air rig Fig.5 driven by a 16" s.w.g. centrifugal fan delivery 620 c.f.m. The main advantage of the air rig is that two-phase flow and cavitation effects were eliminated. The maximum Reynolds number obtainable from this circuit was only half that achieved with the water circuit, but for the majority of the tests the loss coefficient had become independent of Reynolds numbers before this limit had been reached. For those tests where the loss coefficient was still Reynolds-number dependent the flow rate range was extended by pressurising the circuit and thus increasing the density of the air.

DISCUSSION OF RESULTS

The performance of the three diodes is defined as the ratio of the resistance coefficient in the reverse-flow direction to the resistance coefficient in the forward-flow direction at the same Reynolds number.

$$\Sigma = \frac{K_{\text{reverse flow}}}{K_{\text{forward flow}}}$$

where the resistance coefficient K is the ratio of the head loss measured across the diode to the mean velocity head in the pipe at entry to the diode.

$$K = \frac{2g\Delta p}{\rho \bar{v}^2}$$

Vortex Diode

These tests were carried out on the water rig Fig.4. The two main objectives being

(a) to determine the performance characteristics of a vortex diode designed for a specific application.

(b) to determine the effect on performance of different static circuit pressures.

Over the test range the performance, Figs. 7, 8, and 9, of the design Fig.6, was found to be independent of Reynolds number and equal to 12.5:1. In view of the increase in efficiency measured by Heim(1) when projecting the axial pipe into the vortex chamber, it was decided to attach a nozzle to the axial pipe projecting it to half the depth of the chamber, (the optimum condition found by Heim(1)). No significant change in resistance was noted in the forward-flow direction Fig.9 whereas in the reverse-flow direction the experimental results show that the resistance coefficient levels off at a lower value than for the original design. It is likely that this is caused by the vortex being of decreased strength due to the effective reduction in the size of the chamber. The results confirm Heim's observation at low flow rates. However his work does not indicate what would happen at high flow rates. On the other hand measurements made by Heim(1) suggest that the smaller the vortex chamber, the less effective the diode becomes, which would appear to support the above reasoning at high flow rates.

To determine the effect on performance of different static circuit pressure, tests were carried out on the original design both with and without the axial nozzle. The results Fig.10 indicate that over the range of static pressures tested, the performance remains independent of static pressure. The only effect observed was that the first appearance of the cavitation core at the centre of the vortex was suppressed by increasing static pressure.

Cavitation shown in Fig.10 did not affect the reverse-flow performance over the range of flows tested, a conclusion confirmed by Heim(1) in his work. On the other hand Heim(1) did find that in the forward direction of flow cavitation in the throat of the tangential port can make a considerable difference to the forward-flow resistance. In this work cavitation was not present in the tangential port during any of these tests.

The poor performance of this particular vortex diode appears to be due mainly to the following factors:-

(a) The vortex chamber to axial port diameter ratio was restricted by the space available to 50% of Zobel's optimum design.

(b) The maximum velocity through the diode was limited to 40ft/sec because of cavitation considerations.

(c) It was necessary for the flow to pass through a right angled bend close to the axial port Fig.6, again because of the limited space available.

Scroll Diode

Tests on the original design Fig.12 for At/Ai = 0.324 gave a disappointing performance Fig.13, the best measured value was E-1.7. Observations of flow patterns obtained by dye injection showed that the jet leaving the throat of the venturi was not penetrating sufficiently into the cup to set up the required swirl.

Some improvement was observed when the cup and Venturi were brought into closer proximity. This observation was later confirmed quantitatively by improvements in performance Fig.13.

In order to improve further this penetration other models were made with smaller venturi throat diameters to increase the velocity of the jet. It was also thought that an annular forward-directed step suggested by Linderoth (3) would make the flow separate from the wall and be directed towards the axis of the cup in the form of a free jet.

A limited number of tests carried out to investigate the effect of this step with varying height showed only a small improvement in performance.

The centre bar which was incorporated in the first model was removed from all subsequent models so that the shape of the scroll section agreed more closely with Linderoth's design.Fig.14.

The above modifications led to the testing of six variations of the basic design Fig.15.

For all these six designs the results indicated in general that for Reynolds numbers greater than 5 x 10^4 the performance becomes independent of Reynolds number over the range of flow rates tested. In the particular examples shown in Figs.16,17 & 18 this independence is observed to occur above a Reynolds number of 3 x 10^4.

The results plotted in Figs.13, 19 & 20 are the values of loss coefficient for Reynolds numbers greater than 5 x 10^4.

The main characteristic of the K_R and K_F v Ax/Ai curves for constant At/Ai Figs.19 & 20, is that at first they fall rapidly with increasing Ax/Ai but tend to level off to constant value for Ax/Ai greater than 0.25. The overall result is that the loss coefficient becomes independent of the spacing between the cup and venturi for relatively large values of this distance.

The effect of other geometrical changes, e.g. removal of the venturi step section Fig.15e also affected the flow resistance.

In this particular example the value of K_F has been reduced below that of design Fig.15c owing to a more rounded entrace to the throat even though the throat diameter is smaller. The conclusion is that the performance of this diode is very sensitive to small changes in profile in the region of the venturi step section.

In Fig.13, three of the curves show the same general shape and the indications are that the remaining three, each of which is formed by only two points, follow similar trends.

The maximum performance for each design occurs in the region of Ax/Ai = 0.08. Therefore the position of the cup relative to the venturi for peak performance is very nearly independent of the design changes introduced in these experiments.

The best design tested at present Fig.15e gives a maximum performance of 6.6:1 for At/Ai = 0.045.

For these particular experiments a complete correlation of the results was not possible because several of the parameters were changed in each of the six designs, a situation brought about by the requirements of the specific application. A systematic study of the effects on performance of changes in each parameter is still required. Some unpublished work has been carried out by George (5) on a wedge shaped section of the scroll diode in which flow patterns could be studied with ease through the perspex walls of the model. The results from this work are in general agreement with those obtained from the axi-symmetrical design, and further, the flow patterns at the maximum obtained performance are very similar to those sketched by Linderoth (3) in his paper.

The conclusion is that the best reverse to forward flow pressure drop obtainable from this type of diode will of the order of 7-10:1.

FLUID FLOW RECTIFIER

The most important parameters influencing the performance of this diode are:-

(a) The pitch and number of blades per cascade
(b) The number of cascades
(c) The helix angle of the cascades
(d) The blade profile
(e) The blade angles
(f) The overall diameter of the diode
(g) The fluid properties.

In the present investigation the effects of varying blade profile and number of cascades were not studied. The remaining parameters therefore lead to the following dimensionless groups.

$$\frac{\Delta P}{\rho \bar{v}^2} = f\left(\frac{\rho \bar{v} d}{\mu}, \beta, \theta, L/d, P/d\right)$$

(See notation)

where $\frac{\rho \bar{v} d}{\mu}$ is the Reynolds Number

and $\frac{\Delta P}{\rho \bar{v}^2}$ is proportional to the Resistance Coefficient Factor.

The procedure adopted for this design Figs.21,22,& 23 was to determine the effect of one dimensionless group on performance whilst keeping the remaining groups constant.

For the majority of these tests the closed circuit air rig was used although to observe flow patterns some tests were carried out with the water rig.

Figs.24,25,26 are typical curves for K_F, K_R and Σ v. Reynolds number.

In the forward-flow direction K_F would appear to become independent of Reynolds number above $Re = 6 \times 10^4$. In the reverse-flow direction the point at which the loss coefficient appears to become independent of Reynolds number increases with length from $Re=4\times10^4$ for $L/d = 1.36$ to $Re=12\times10^4$ for $L/d = 2.6$.

Over the range tested (helix angle=60°) the forward-loss coefficient was found to be independent of blade pitch Fig.27,whereas the reverse-loss coefficient varies both with blade pitch and cascade length Fig.28. Since K_F is independent of pitch it follows that the performance curves Fig.29 are dependent for their shape on the Kr.v. L/d curves. A reduction in pitch, i.e. a larger number of blades for a given cascade length, results in a greater pressure rise in the reverse-flow direction. This greater pressure rise is reflected in the gradients of the Σ v L/d curves. Similar trends are shown for a larger helix angle ($\theta = 90^\circ$), Figs.27,30 and 31, but for a cascade of given pitch and length both K_F and K_R increase with helix angle over the range tested.

Observation of flow patterns using water clearly showed that small eddies are formed behind each blade in the forward-flow direction. These are the main contributors to this head loss. In the reverse direction of flow the observations confirmed that spinning flow is obtained over 90% of the cascade length.

The pitch/diameter ratio Fig.32 for maximum performance is approximately 0.11 for a helix angle of 60° and this value is independent of the number of blades per cascade. A comparison of Fig.32 and 33 shows that the pitch/diameter ratio for best performance has decreased with a larger helix angle.

The effect of increasing the helix angle of the cascades from -60° to $+90^\circ$ is to considerably improve the overall performance of this design Fig.34. This is brought about mainly by the improvement in the reverse loss coefficient. The greater the helix angle, the less the obstruction of the leading edge of each subsequent blade. This permits the blades to impart circumferential flow to the fluid much more readily than with the blades located directly behind one another.

On the other hand, the passages between the cascades have the same helix angle, so that the forward-flow head loss will also increase with increasing angle. It can be expected therefore that there will be a critical helix angle beyond which the performance will begin to fall, because the forward-flow head loss will become the predominant factor.

For any given number of blades tested there is a peak performance which occurs at a value of Pitch/diameter ratio of approximately 0.11, Figs. 32 and 33. This value is unaltered by a change in helix angle from + 60° to + 90°

The effect of increasing the number of leading blades with varying angles Fig. 35 is to reduce the overall performance for short lengths of cascades. For long cascades the performance is unaffected for these particular angles.

In the reverse-flow direction, the blades at right angles to the axis of the pipe contribute a greater pressure rise than the leading blades which are inclined to the axis. Consequently it would be expected that the greater the proportion of right angled blades the higher the loss coefficient.

All test results indicate that the greater the cascade length the better the performance. Inspection of these results suggests that the performance will pass through a maximum as the length is increased. This will be brought about because the increasing loss in the forward-flow direction will, eventually have a predominating influence. It is also expected that this critical length will vary with varying pitch/diameter ratio.

Further investigation will be made on this diode to establish the optimum helix angle and length and also the associated optimum performance.

COMPARISON OF THE THREE DIODES

Table 1 Fig.36 gives the best performance data so far obtained from the three designs.

The results show that the Vortex diode gives the best overall performance and a small forward - flow loss coefficient. The main disadvantage of this design is that for a given flow rate its overall size will be much greater than that of the rectifier and scroll diodes.

Both of these latter designs are geometrically more suitable for fitting into a pipeline, although the rectifier would be more expensive to produce and more liable to mechanical damage by the passage of foreign bodies.

Unfortunately, the scroll diode not only has a poor overall performance but could be expensive to operate as the forward flow loss coefficient is extremely high.

In some applications a relatively high pressure drop in the forward-flow direction can be tolerated in order to achieve a very high pressure drop in the reverse direction. The scroll diode would fit this requirement, although it is important to realise that the addition of flow restrictors to the rectifier and vortex diodes can achieve a similar result, but at the cost of a reduction in their overall performance.

It is thought that the results of further investigations into the optimised design of the rectifier will produce figures comparable with the vortex diode but having the above mentioned advantage of passing much larger quantities of fluid on a size for size comparison.

If a liquid is used, cavitation is less likely to occur in the rectifier because the local fluid velocities are much lower than in the vortex diode.

CONCLUSIONS

The principal conclusions to be drawn from the experimental investigations are as follows:-

(a) The vortex diode tested gave a best performance of 12.5:1, although it is known that performances as high as 50:1 have been achieved. This considerable disparity was due to limitations imposed by design requirements of the particular application. The low forward-loss coefficient of the vortex diode compared favourably with that of the fluid flow rectifier.

(b) The performance of the scroll diode was not encouraging giving a maximum ratio of only 6.6:1. It also has a very high forward-loss coefficient which would be undesirable for most applications.

(c) The fluid flow rectifier gave a best performance of :1 and a relatively small forward-flow head loss. It would probably be more suitable geometrically for most applications than the vortex diode, although more expensive to manufacture.

The work so far carried out shows that further experimental investigations are necessary particularly on the rectifier before optimisation is achieved.

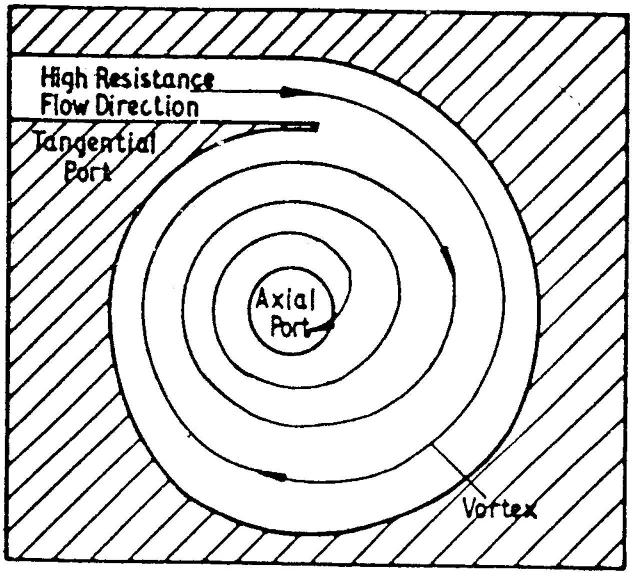


FIG. 1. EXPECTED FLOW PATH FOR THE VORTEX DIODE

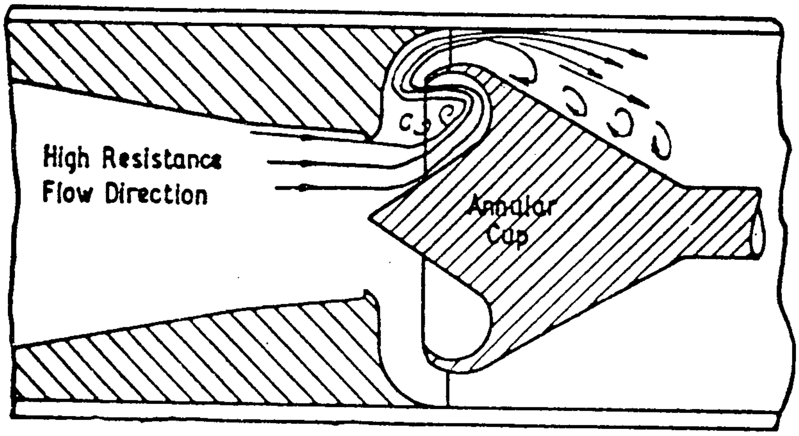


FIG. 2. EXPECTED FLOW PATTERNS FOR THE SCROLL DIODE

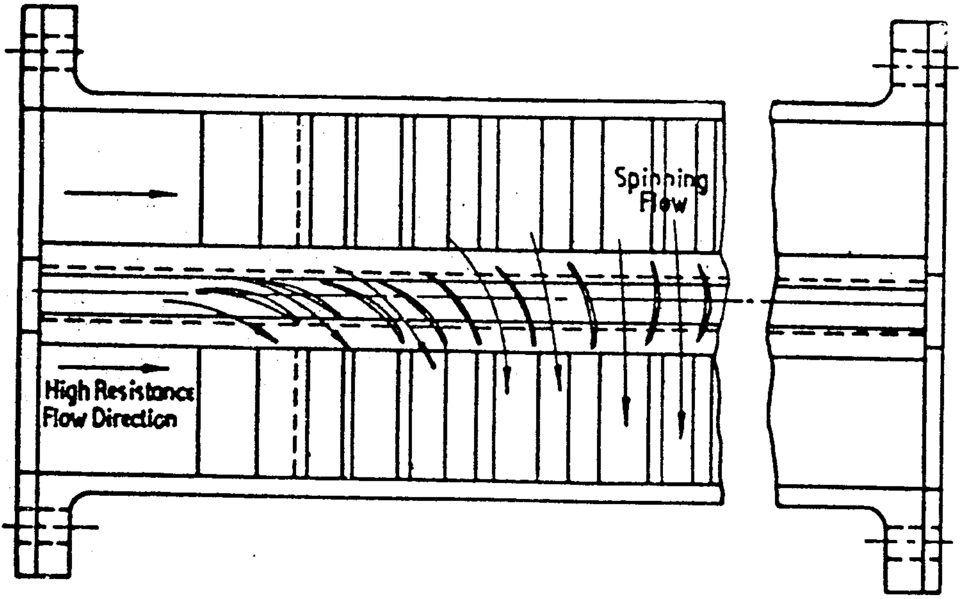


FIG. 3. EXPECTED FLOW PATH FOR THE FLUID FLOW RECTIFIER.

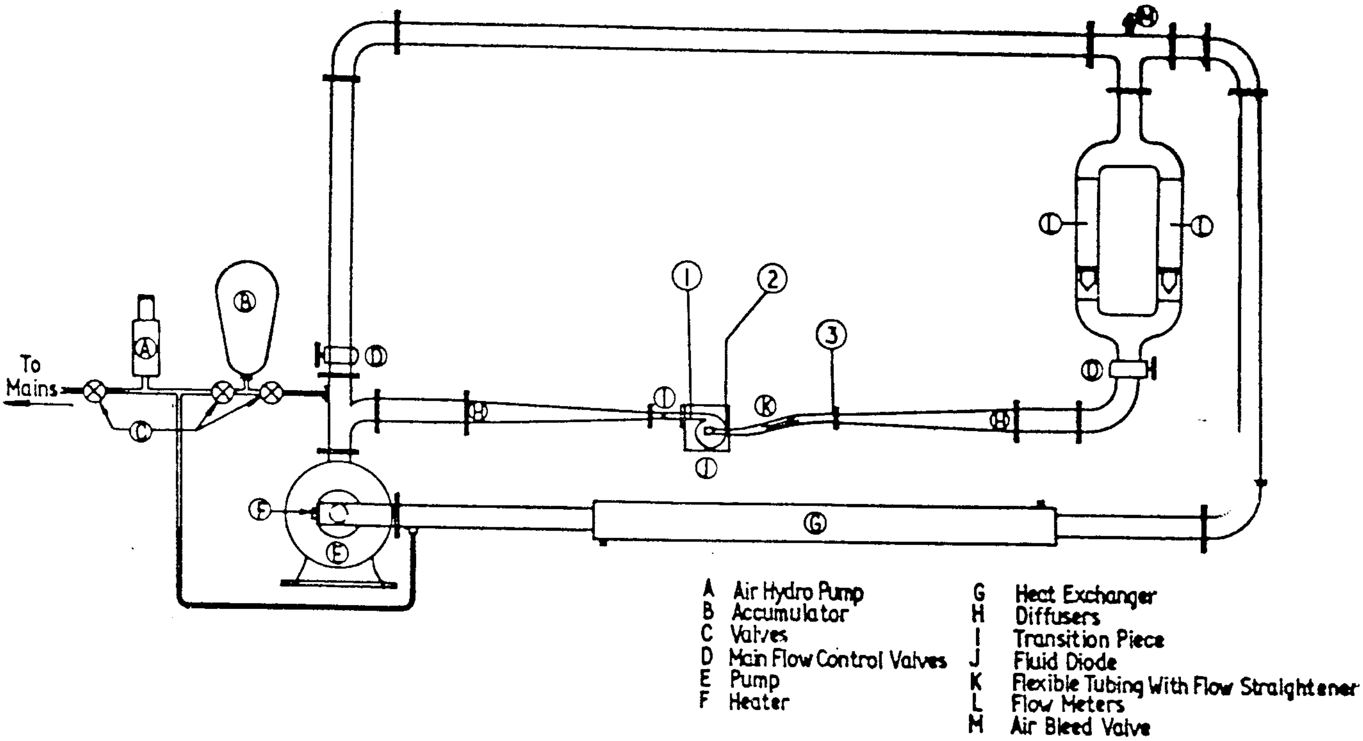


FIG. 4 CLOSED CIRCUIT WATER RIG

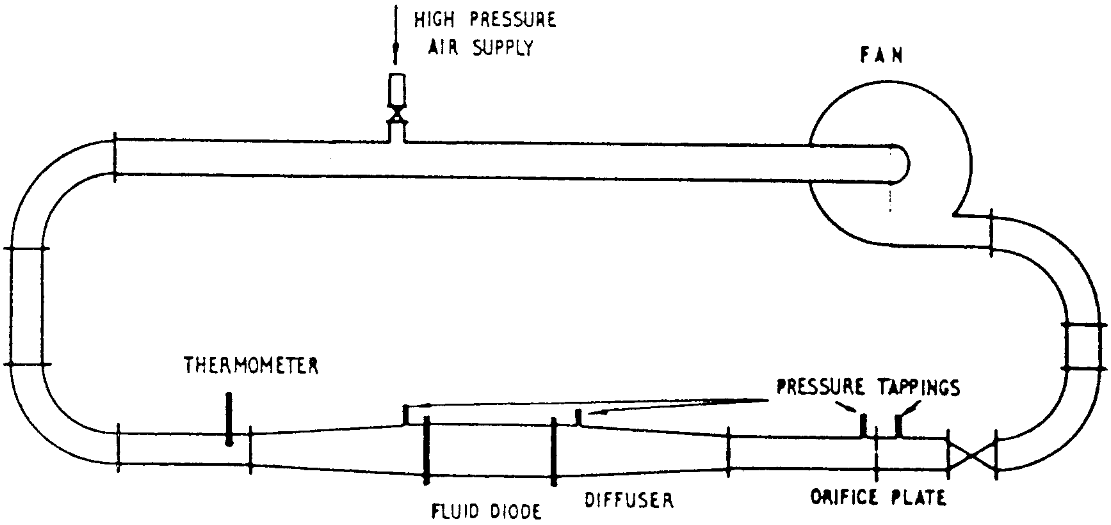


FIG. 5 CLOSED CIRCUIT AIR TEST RIG

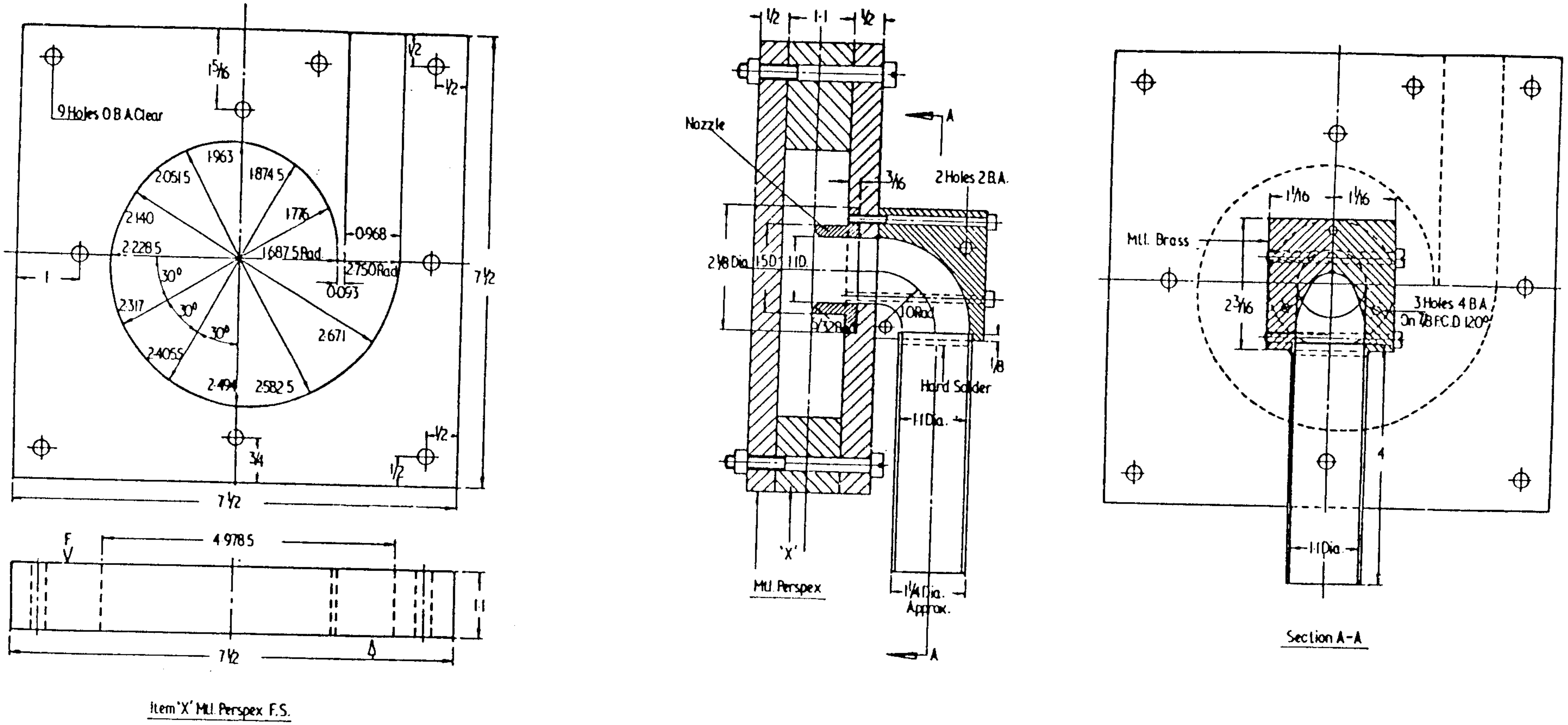


FIG. 6 VORTEX DIODE

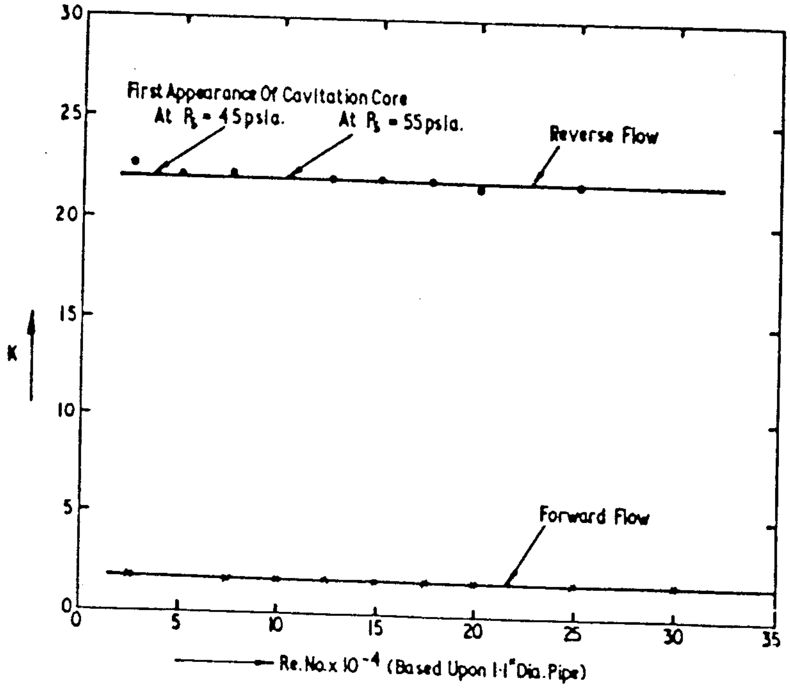


FIG. 7 VORTEX DIODE WITHOUT NOZZLE

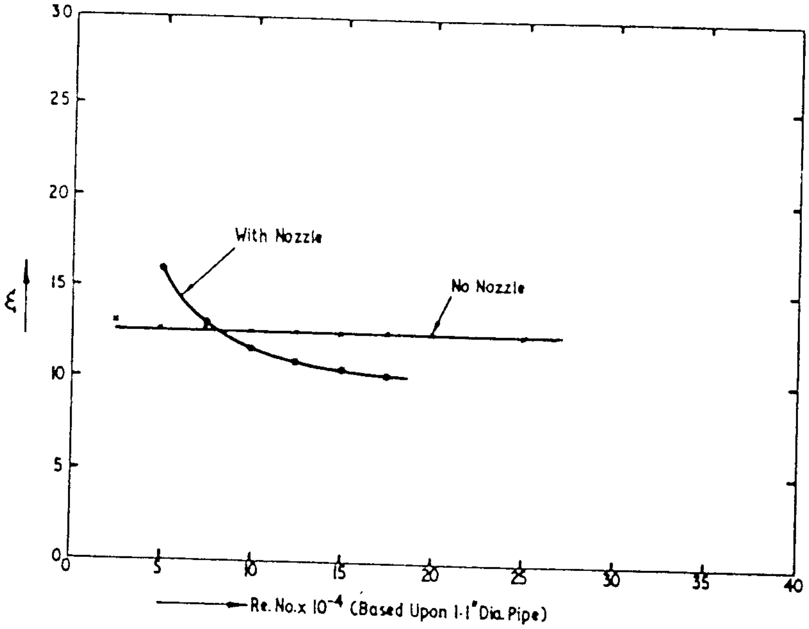


FIG. 8 VORTEX DIODE

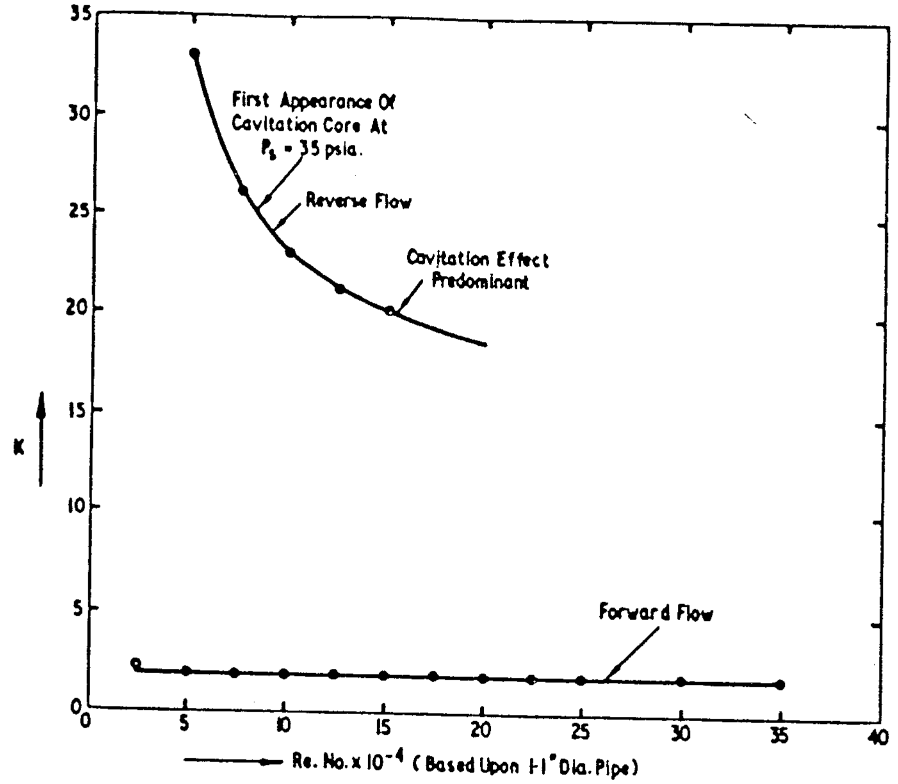


FIG. 9 VORTEX DIODE

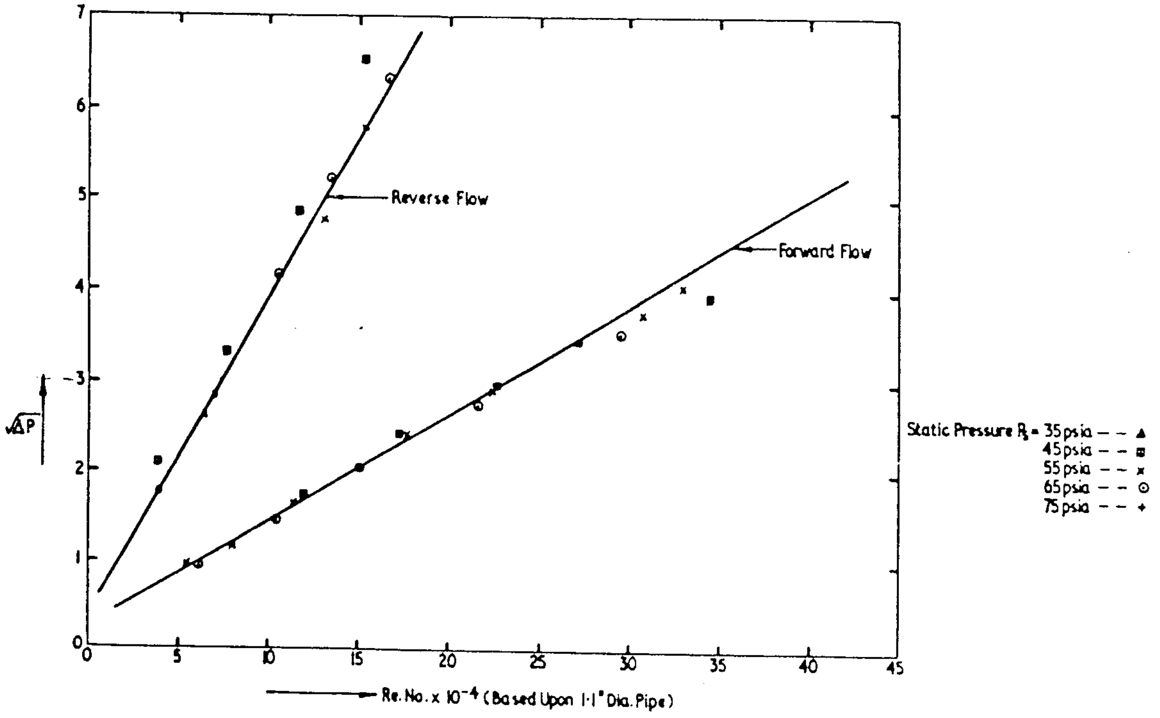


FIG. 10 VORTEX DIODE WITHOUT NOZZLE

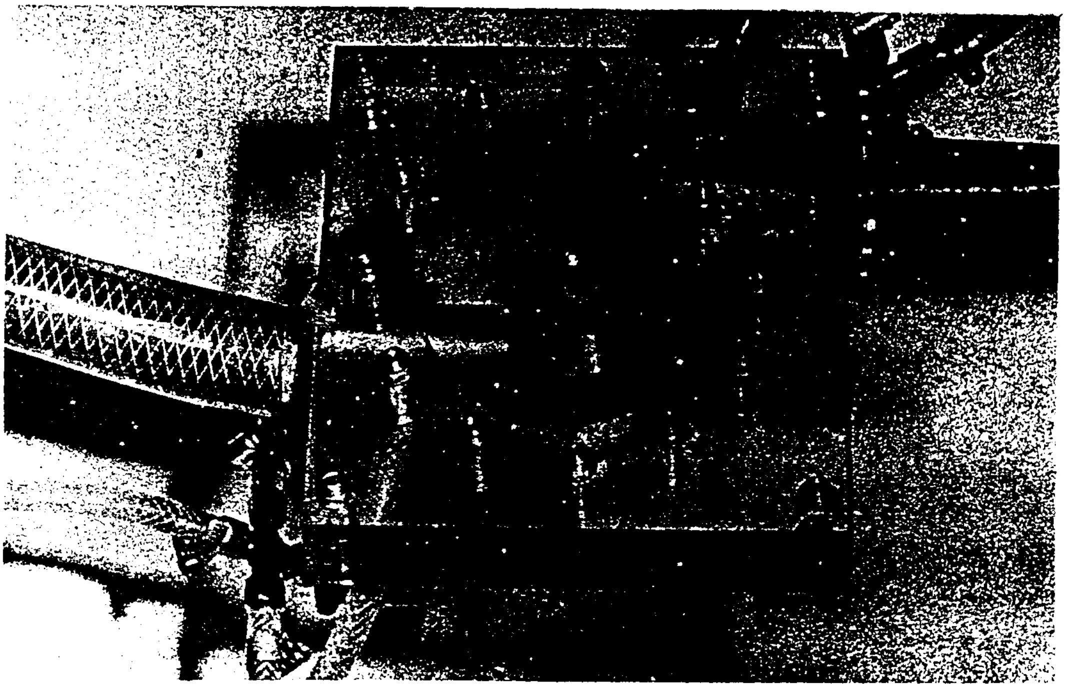


FIG. 11. VORTEX DIODE WITH NOZZLE SHOWING CAVITATION CORE

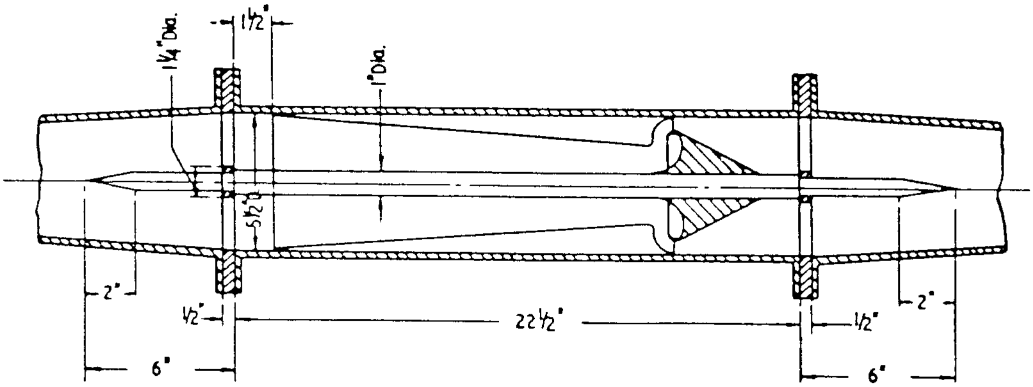


FIG. 12a ASSEMBLY OF SCROLL DIODE IN $5\frac{1}{2}$" DIAMETER TUBE

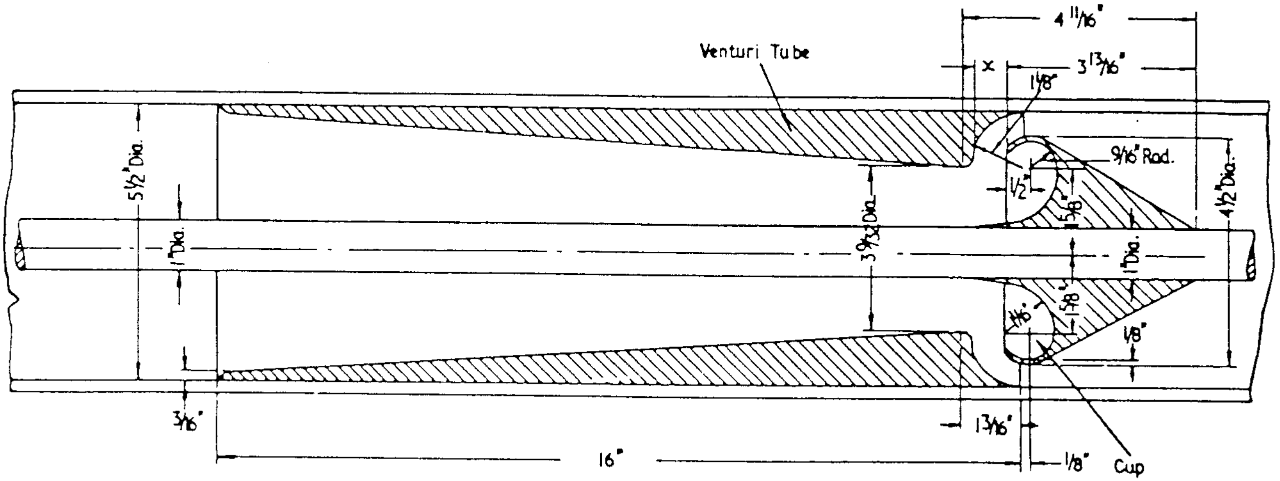


FIG. 12b ORIGINAL DESIGN OF SCROLL DIODE (Not to scale)

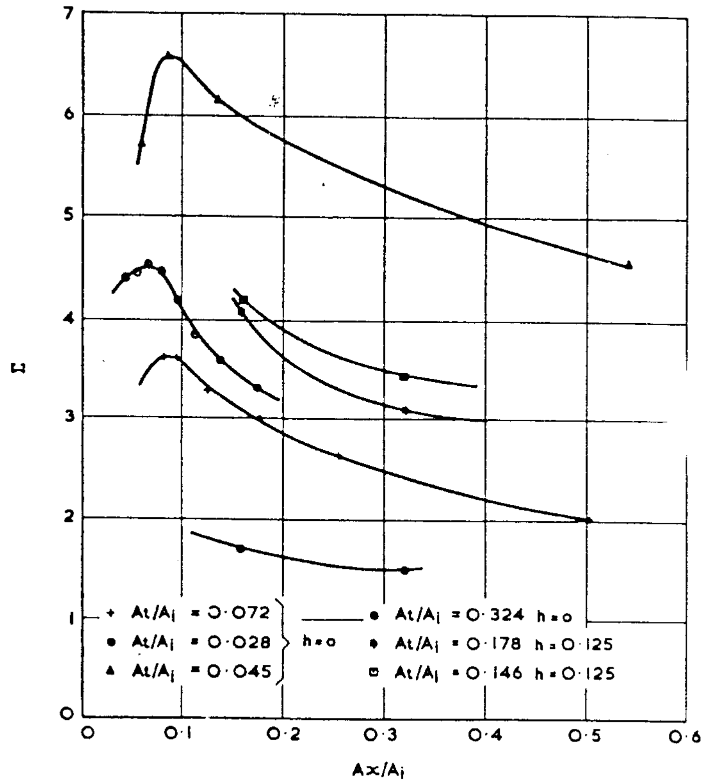


FIG. 13 SCROLL DIODE

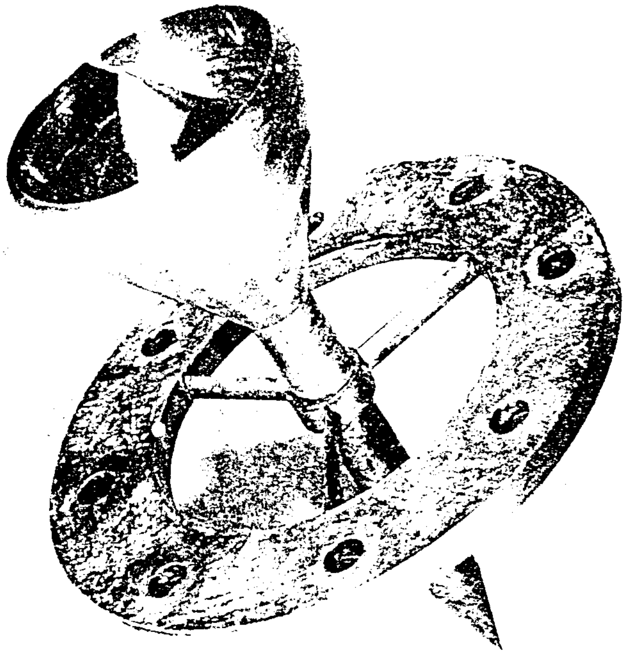


FIG. 14. MODIFIED CUP OF SCROLL DIODE

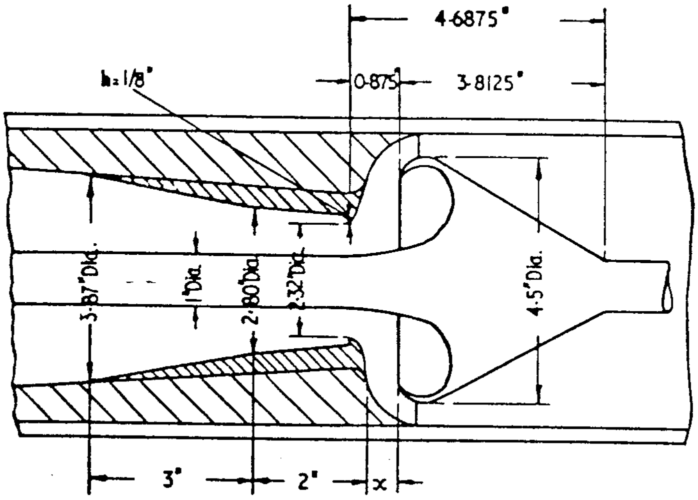


FIG. 15a. MODIFICATION 1 REDUCED THROAT

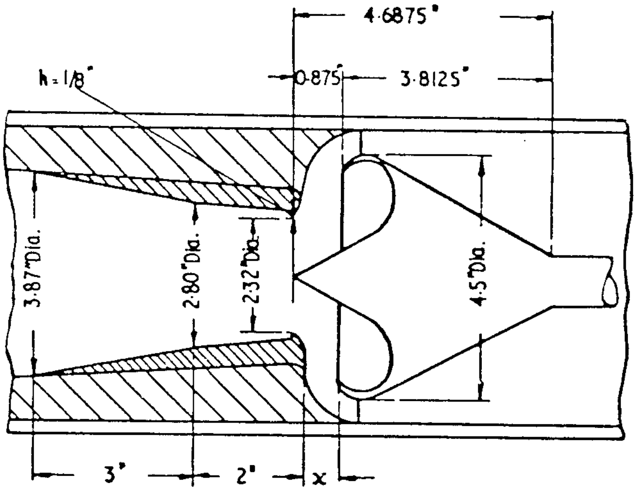
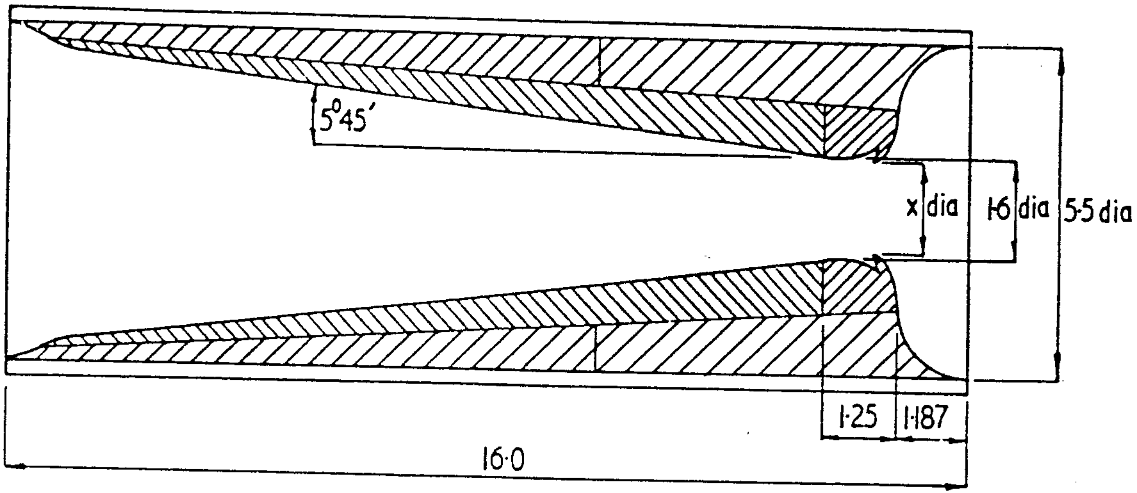


FIG. 15b. MODIFICATION 2 CENTRE BAR REMOVED



x = 1.48" Dia.
1.40" Dia.
1.34" Dia.

FIG. 15c MODIFICATION 3

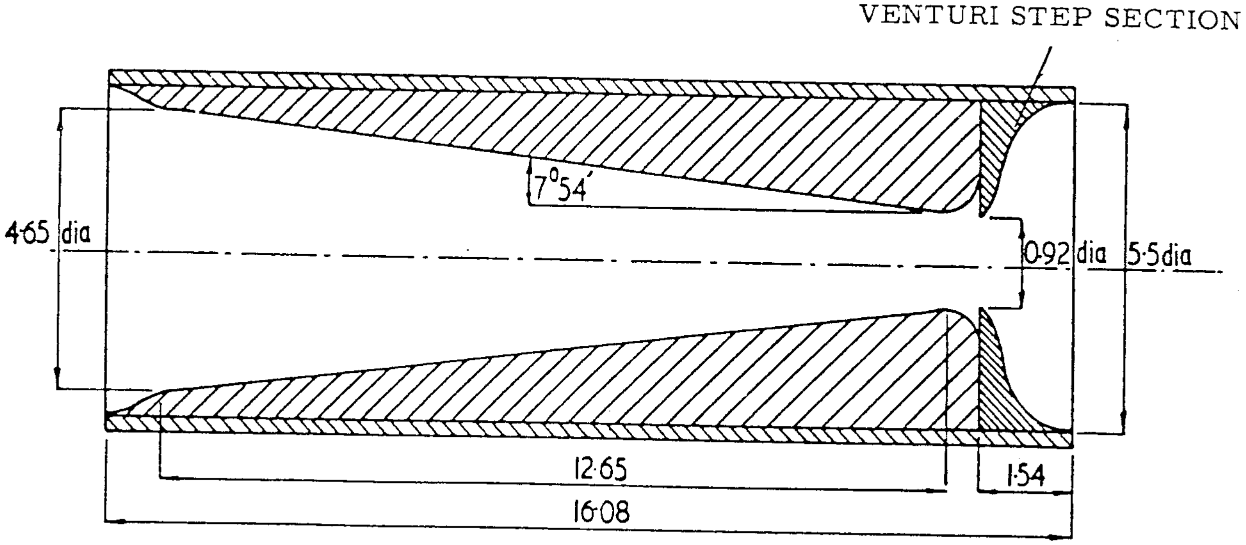


FIG. 15d MODIFICATION 4

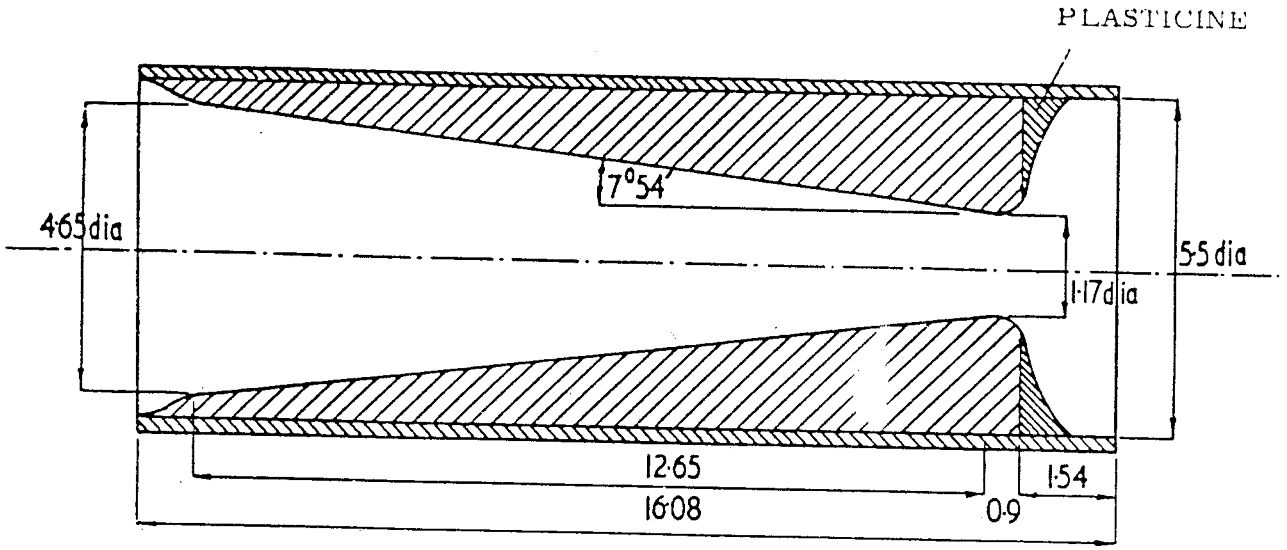


FIG. 15e MODIFICATION 5

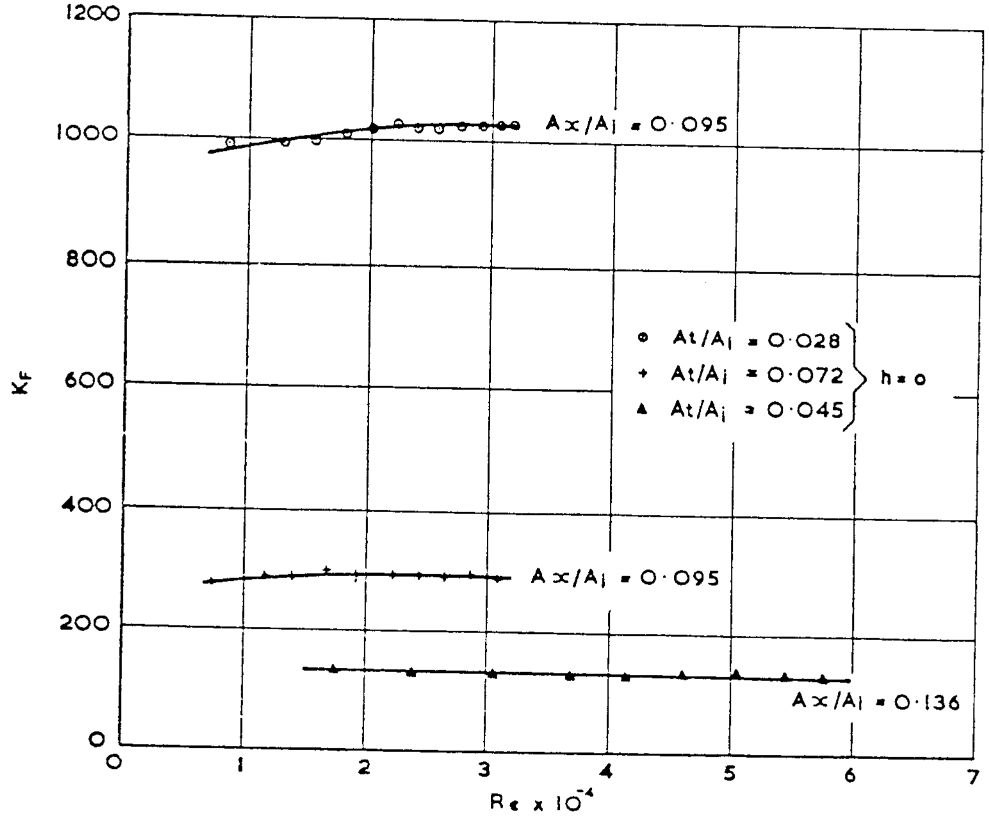


FIG. 16 SCROLL DIODE

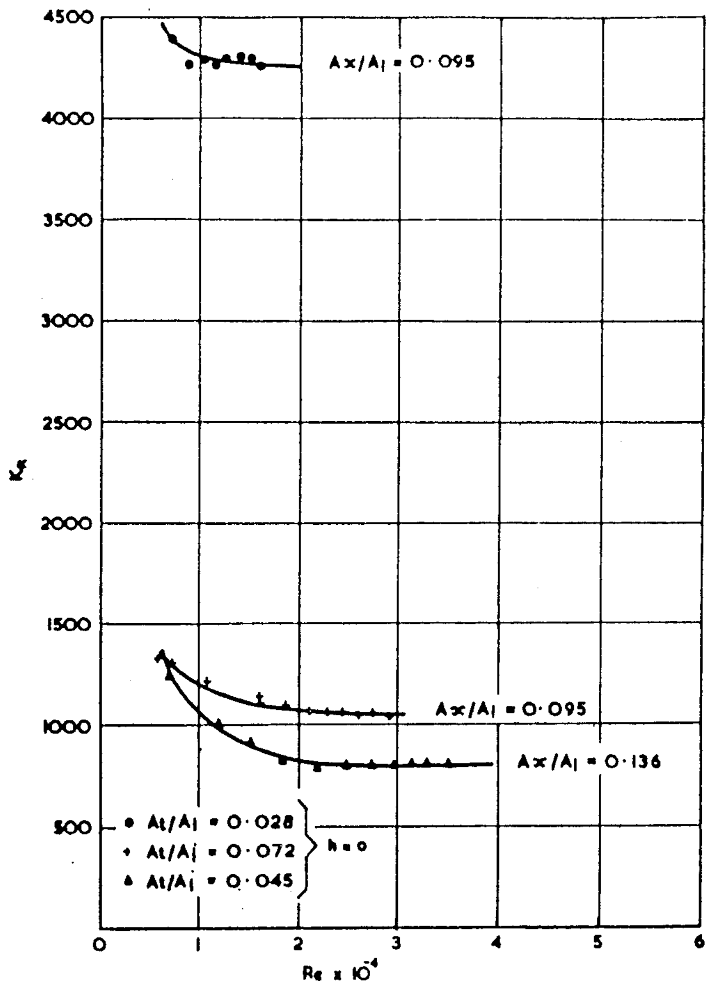


FIG. 17 SCROLL DIODE

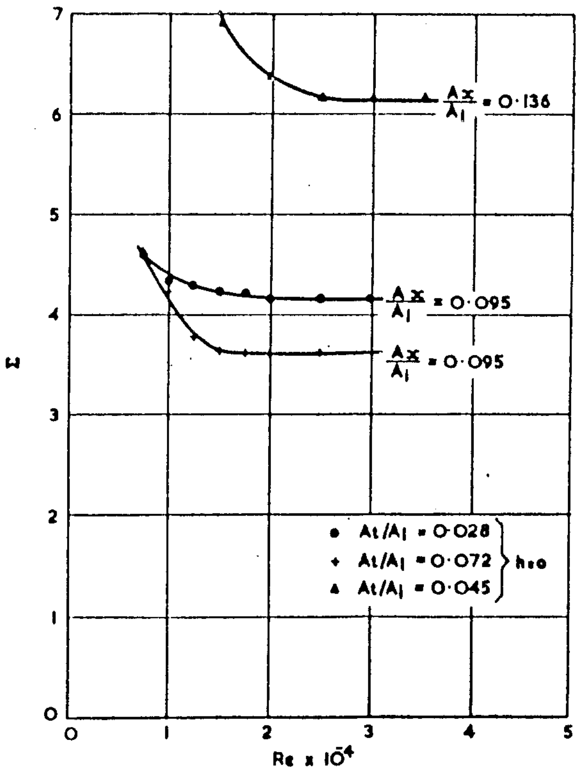


FIG. 18 SCROLL DIODE

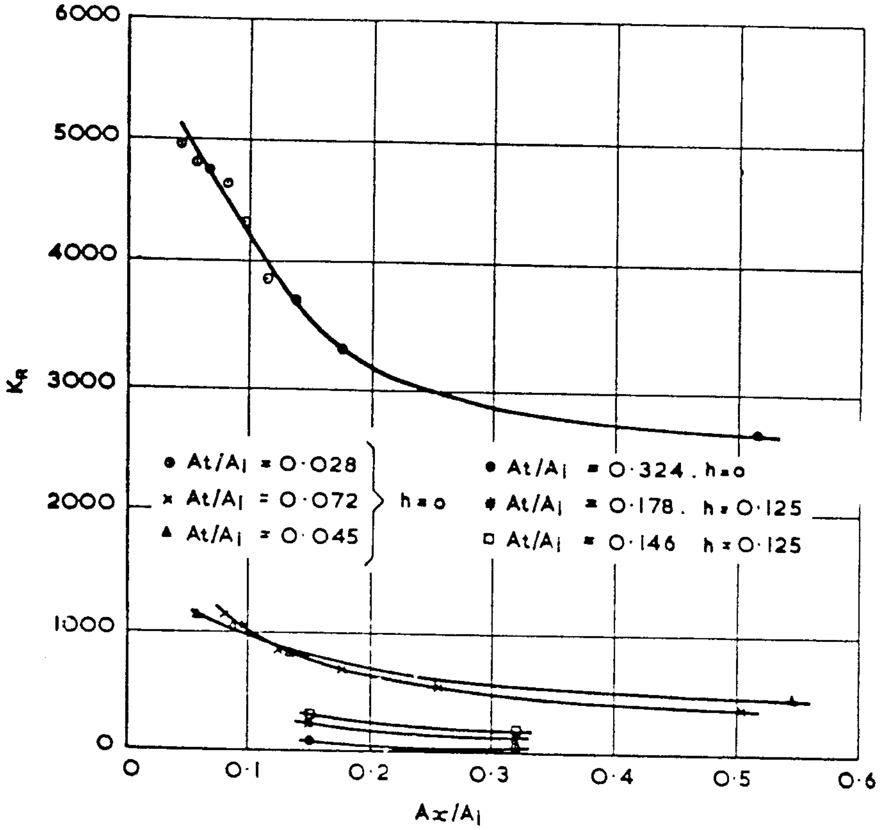


FIG. 19 SCROLL DIODE

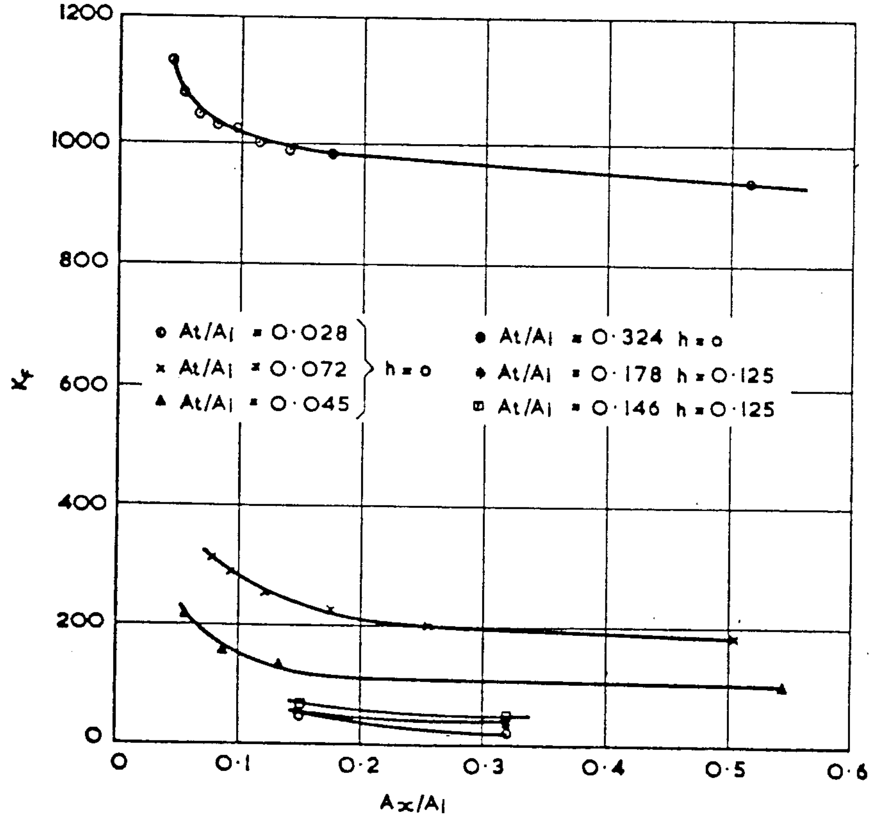
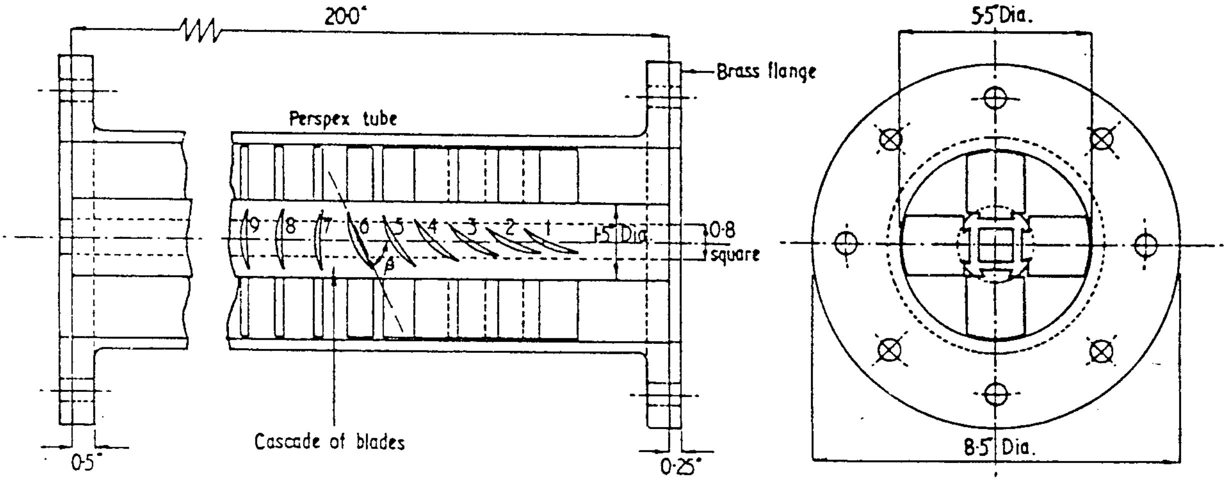


FIG. 20 SCROLL DIODE

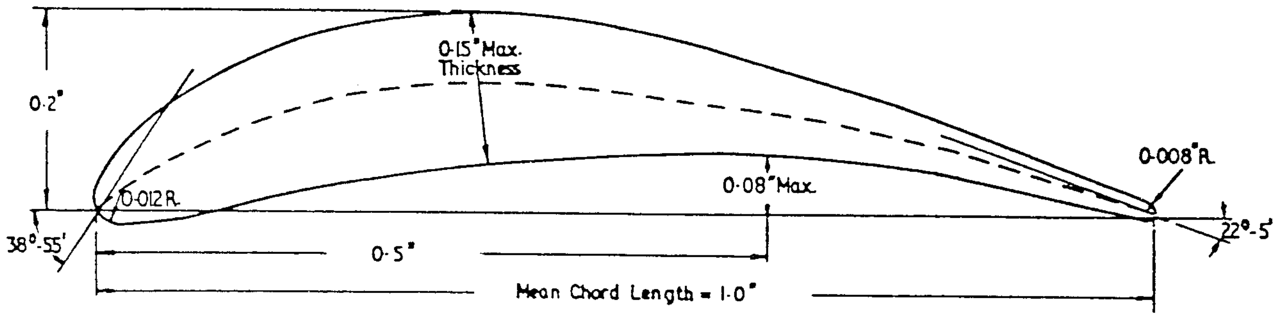


Leading blade angles (β)

Blade	β
1	12°
2	18°
3	30°
4	45°
5	60°
6	73°
7	90°

Diode shown with straight cascade (θ=0)

FIG. 21 FLUID FLOW RECTIFIER



Not To Scale.

FIG. 22 BLADE PROFILE

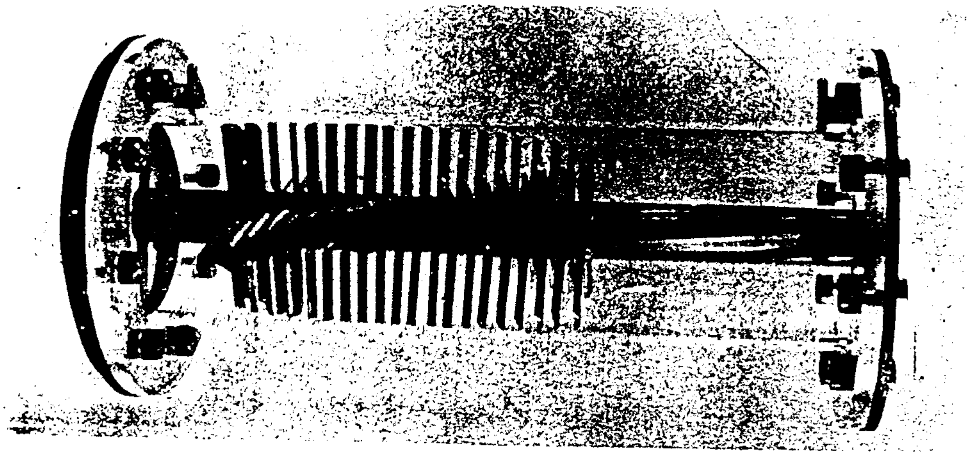


FIG. 23. FLUID FLOW RECTIFIER (HELIX ANGLE 90°)

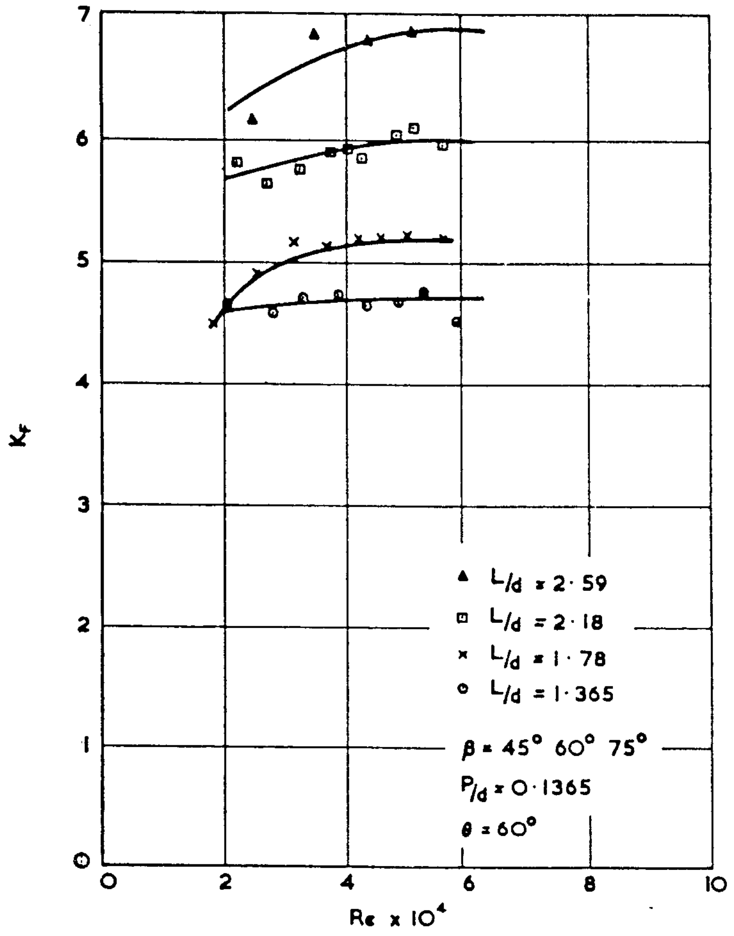


FIG. 24 FLUID RECTIFIER

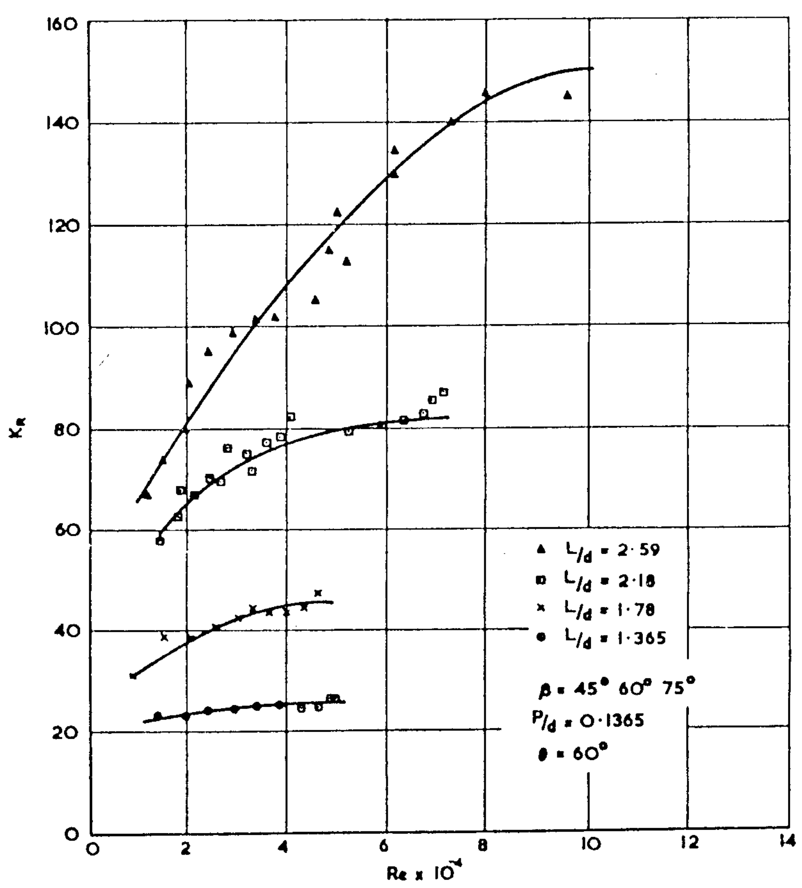


FIG. 25 FLUID RECTIFIER

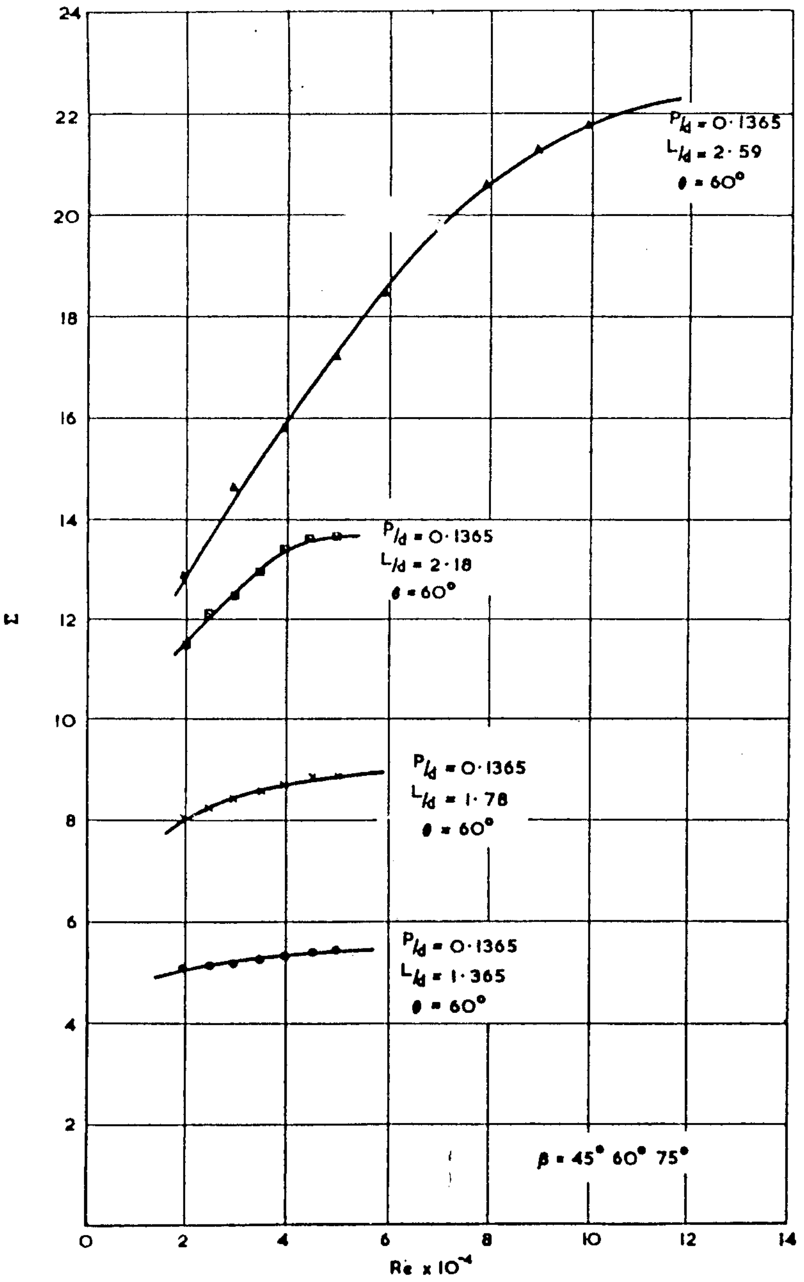


FIG. 26 FLUID RECTIFIER

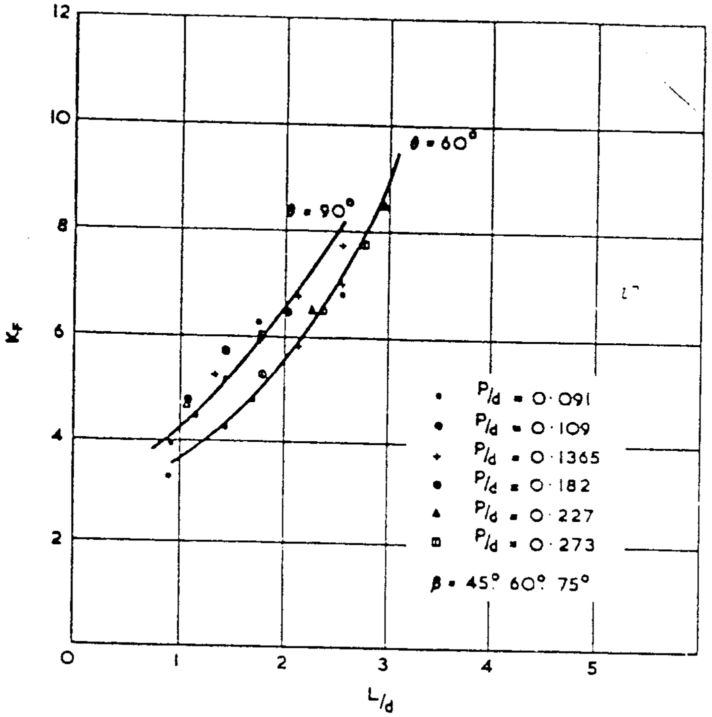


FIG. 27 FLUID RECTIFER

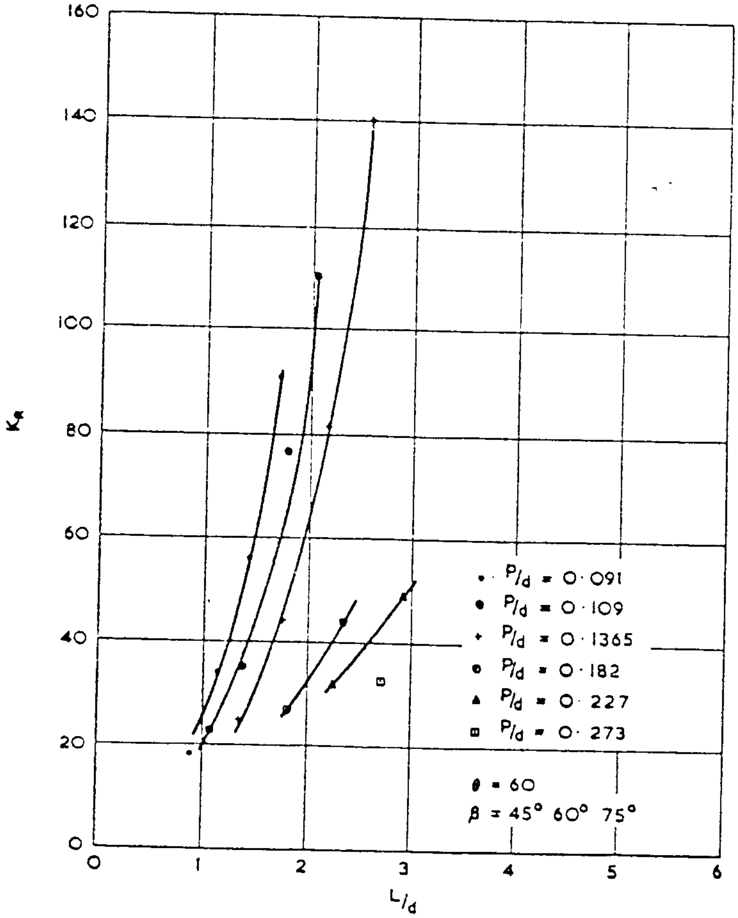


FIG. 28 FLUID RECTIFIER

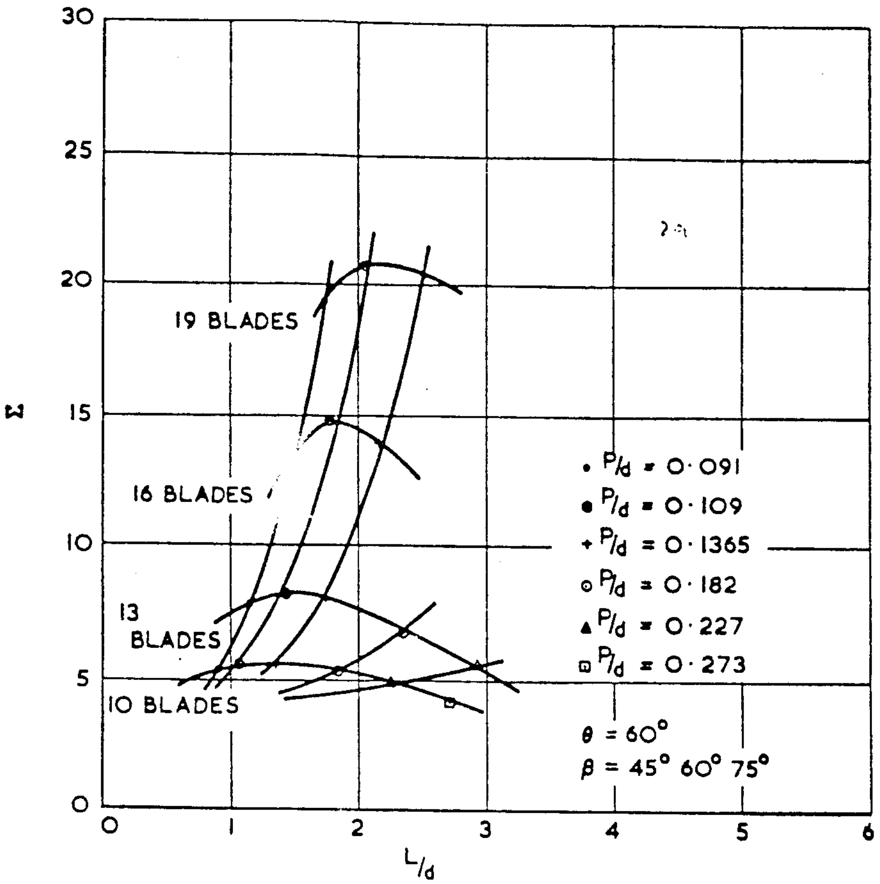


FIG. 29 FLUID RECTIFIER

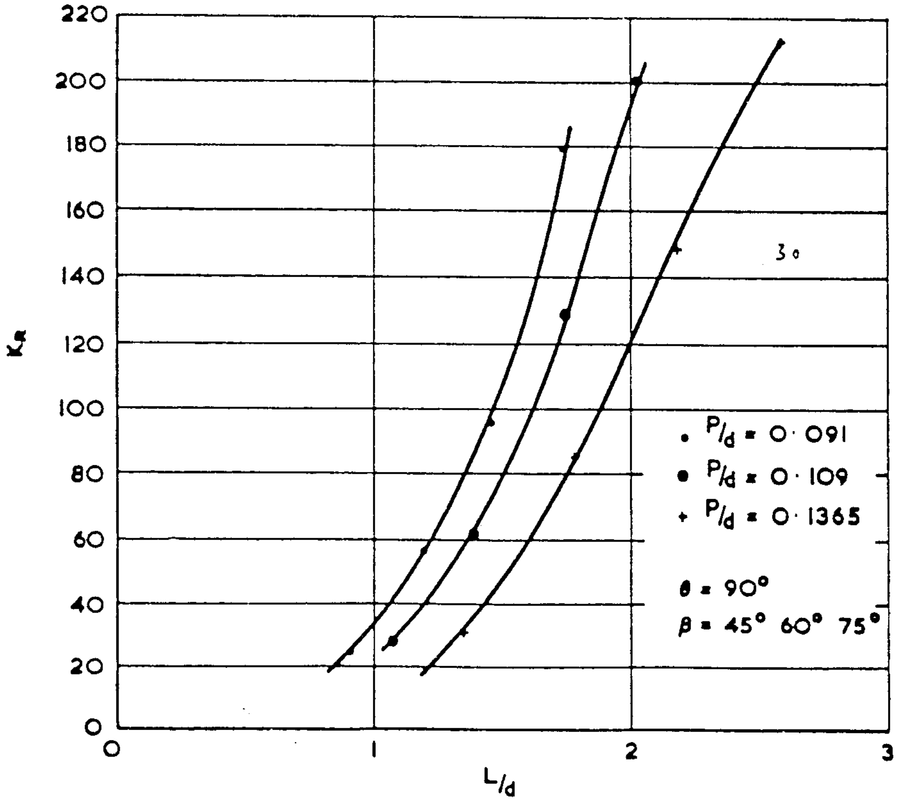


FIG. 30 FLUID RECTIFIER

15c

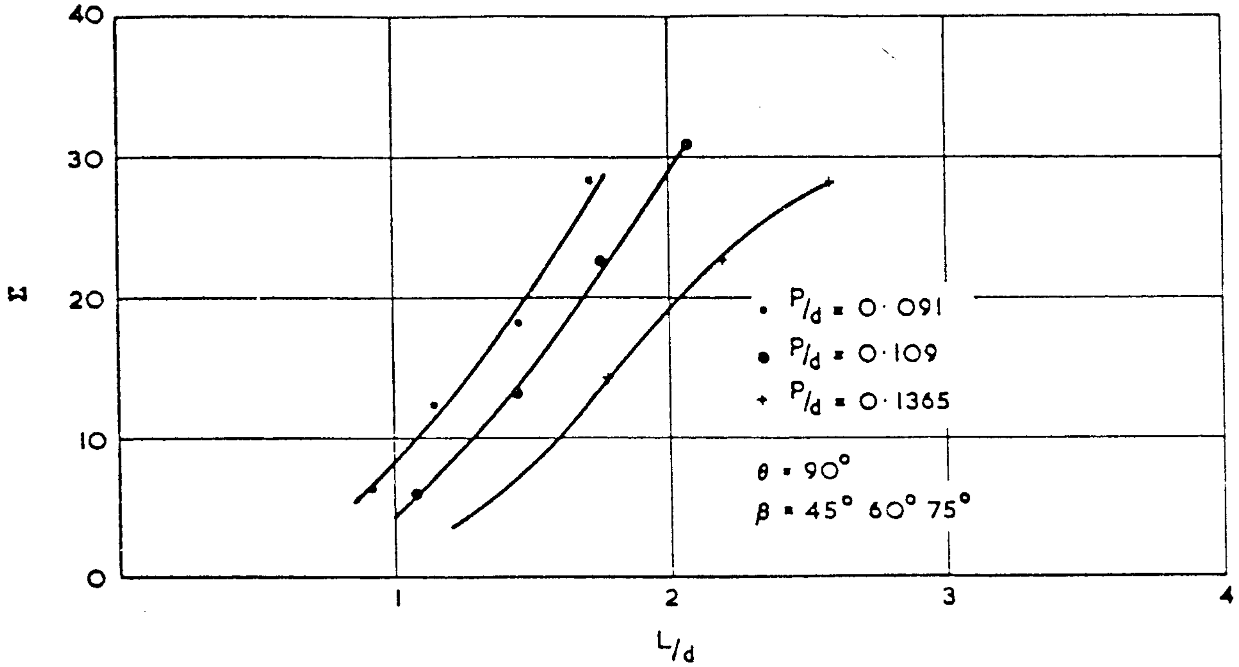


FIG. 31 FLUID RECTIFIER

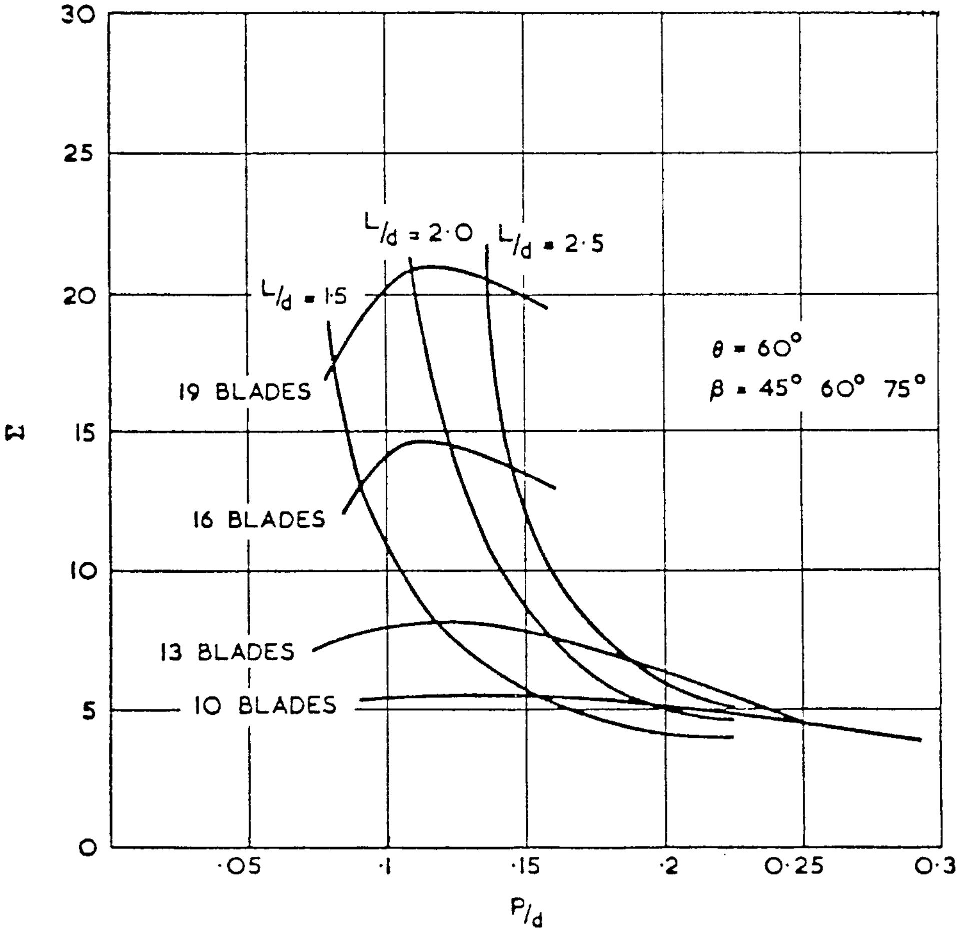


FIG. 32 FLUID RECTIFIER

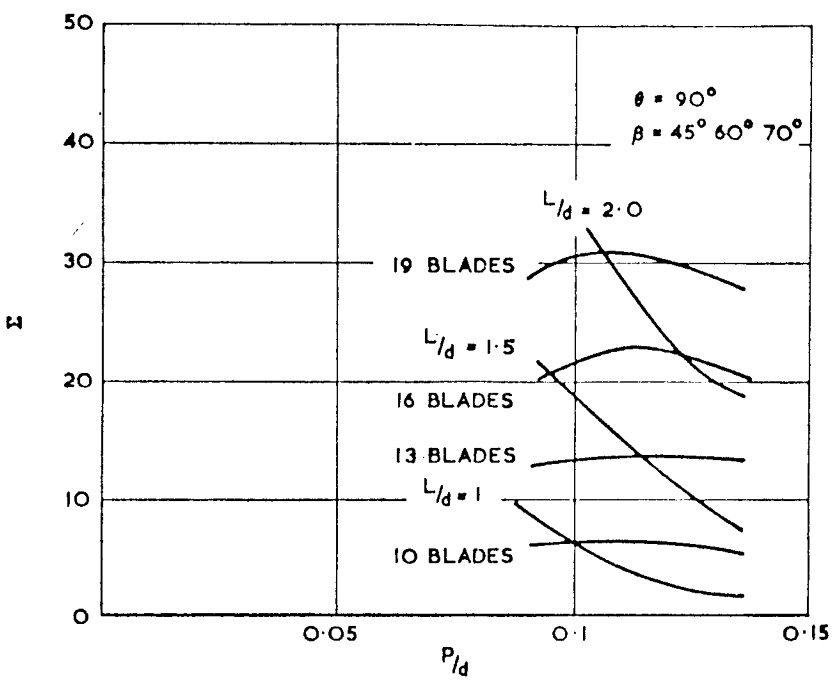


FIG. 33 FLUID RECTIFIER

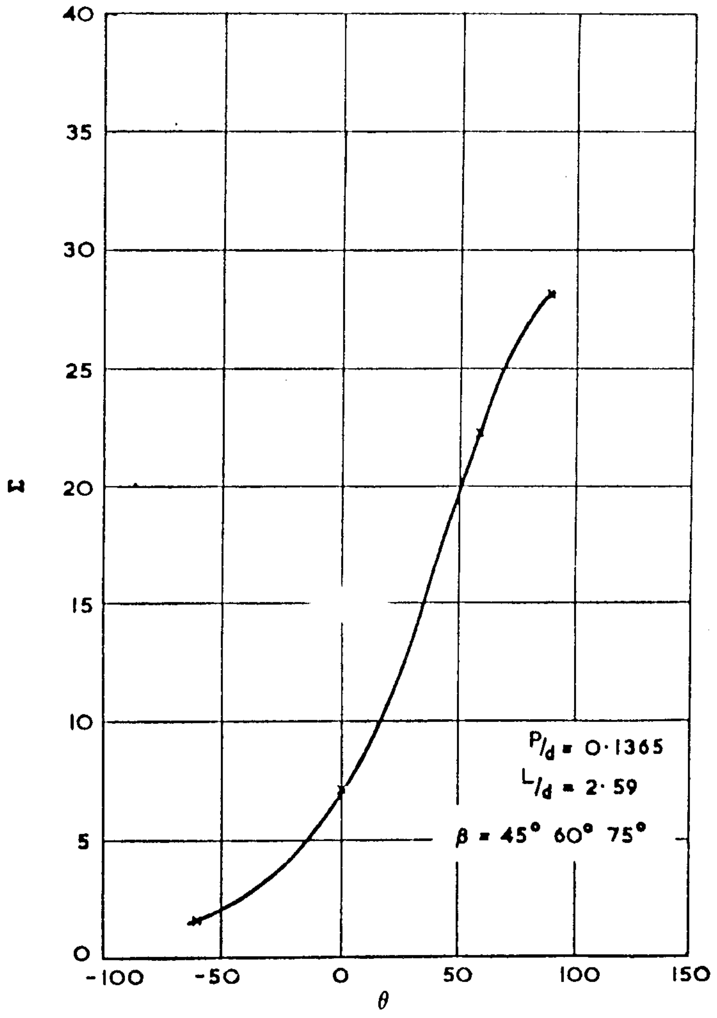


FIG. 34 FLUID RECTIFIER

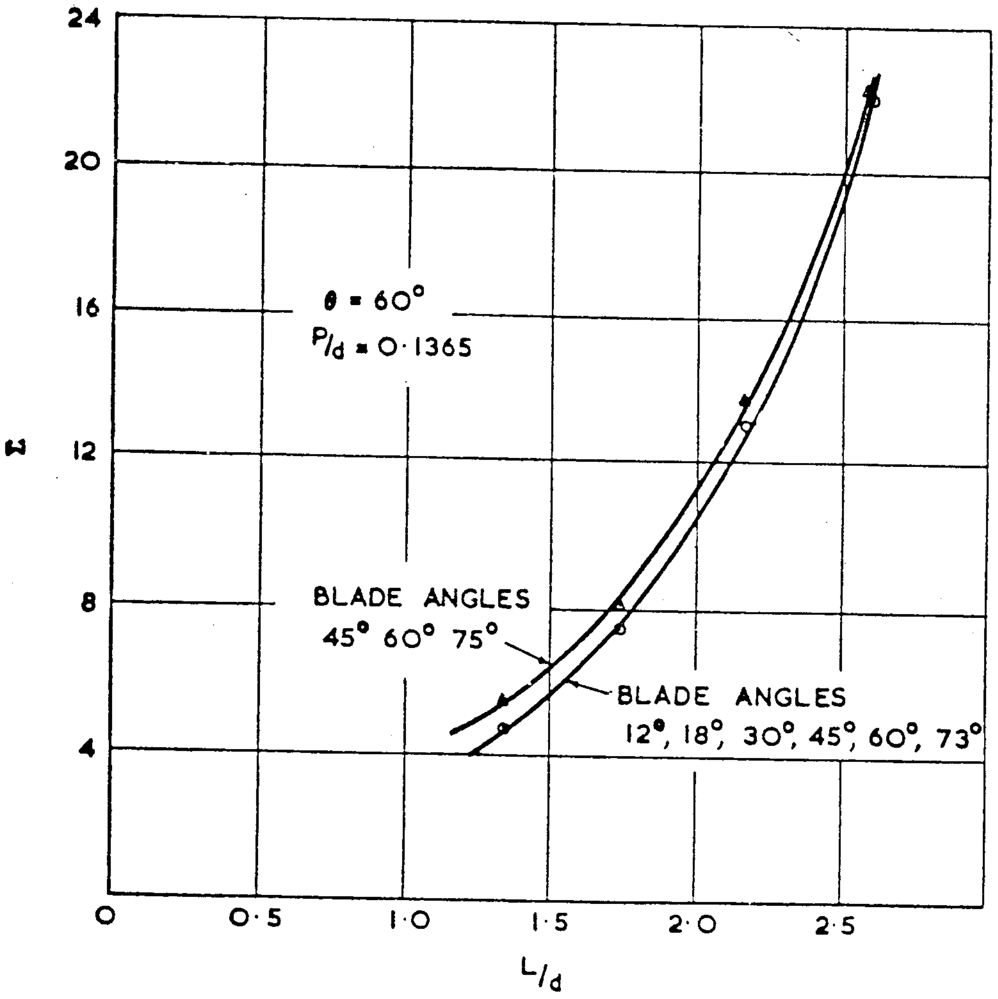


FIG. 35 FLUID RECTIFIER

TABLE 1

	K_F	K_R	MAXIMUM MEASURED Σ
VORTEX DIODE	8	400	50
SCROLL DIODE	152	1000	6.6
RECTIFIER	6.45	200	31

FIG. 36 FLUID RECTIFIER

p. D-122

Dr. P.J. Baker communicated the following addition to his paper D6

A point I would like to make with this paper is that the three designs chosen were for a specific application for the Atomic Energy Authority. The reason why we chose in particular the scroll diode and the fluid rectifier was because both these designs, which incidently are patented, claimed that they were giving extremely high performances of the order of about 100:1. For this reason, these particular designs were chosen.

Another point is that the pure fluid rectifier's maximum performance is recorded in this paper as 31:1. But since this paper has been written we have done some more tests and by extending the number of blades in the length of the rectifier we have in fact obtained about 49:1. There is an indication that the performance is still rising.

p. D-125, 126

GEORGE, P.T. U.K.A.E.A. (Paper D6)

Dr. Baker's paper has discussed steady state performance characteristics of 3 diodes. It would be of interest to know whether any data is available in transient performance especially in view of the devices. Depending on the transient behaviour one can consider using such devices as a safety feature is a reactor to prevent spread of fission product gas from failed fuel cans reaching other channels by blow-back through the entry plenum.

p. D-131

Dr. P.J. Baker replying to Mr. P.T. George on Paper D6

A comprehensive survey of the literature on fluid diodes has been carried out by the information department of B.H.R.A., but there does not appear to be any information available on their transient behaviour. It is proposed to investigate the transient performance of fluid diodes at the B.H.R.A. laboratories some time in the near future. But the general feeling is that the transient performance will be substancially worse than that under steady state conditions.

Coal burns best in pipes that hum

A coal-burning system with a central tubal chamber that resonates like an organ pipe during combustion has been designed by a Georgia engineer. The system's 70-hertz hum is no accident, but instead a design feature that should catapult its energy-conversion efficiency well above the norm, says Ben Zinn of the Georgia Institute of Technology in Atlanta.

There are two goals in optimizing combustion efficiency: to burn fuel as completely as possible, and to burn it with as little air as possible. Incomplete combustion obviously wastes fuel; less obvious is the fact that use of too much air robs the system of heat. The acoustic waves resonating through the prototype "pulsating combustor" make it possible for Zinn to obtain virtually complete combustion with almost no "excess air." Says Zinn, "I'm not aware of anyone ever getting the results we have. I think we have something unique."

Burning can only take place in the presence of oxygen. Just as an auto engine must breathe in a certain critical ratio of air along with its fuel, so must any other combustion system. Theoretically, the amount of air required to burn a given amount of fuel completely is usually not enough, in practice, to prevent the production of smoke. As a result, combustion engineers must always budget in a certain amount of excess air. The *Dictionary of Energy* (Schocken Books, New York 1983) cites as typical values "50 percent excess air for coal-fired [combustion units], 20 percent for oil-fired and 10 percent for gas-fired installations." Any excess air in the system will be heated by the hot combustion process and eventually be exhausted along with other waste gases. The more excess air used, the more heat robbed from the system.

Zinn can get 92 percent combustion efficiency — a figure many energy managers could live with — using no excess air. By adding six or seven percent excess air, he achieves greater than 97 percent combustion efficiency, a value electric utilities strive for with their better systems. Moreover, Zinn points out, to achieve comparable efficiencies, most of those other systems require use of "at least 20 percent and sometimes 30 percent extra air. That," he emphasizes, "is a huge difference."

Zinn's system taps an acoustic principle formulated during the 19th century by a physicist named Rijke: By heating gases within a tube at a critical point, the resulting excitation of gas molecules will generate acoustic oscillations that make the pipe sing. The heat source in Zinn's device is the combustion process itself, which occurs on a porous metal grid inside the pipe. Fuel entering the chamber from a portal along one wall drops onto the grid where it meets cold air that's been pumped in from below.

In his model, the combustion tube is 9 feet long and 5.5 inches in diameter. Its dimensions determine the oscillation frequency, which for this system "means that the molecules in each direction are moving back and forth 70 times a second," Zinn says. "I have such superior combustion because the acoustics give me much better mixing of fuel and air." But the oscillations have a second benefit: It turns out they increase the transfer of heat from the hot combustion-exhaust gases to the walls of the combustion chamber, increasing the energy available to do work — for example to heat the steam that drives a turbine to generate electricity.

"The implication of these advantages is that for the same energy output, pulsating combustors can be smaller than conventional ones and therefore less expensive," Zinn says. Another advantage of his system is that it doesn't require use of pulverized coal, a slightly more costly form of the fuel. Georgia Tech has a patent pending on Zinn's design. —*J. Raloff*

Why is pulsed combustion more efficient?

Pulse combustors, for reasons not fully understood, are more efficient and produce less pollution than conventional combustors of natural gas. Even with the present, rather crude 'state of the art', pulse combustors are between 33% and 70% more efficient than conventional combustors in terms of the percentage of stored chemical energy that converts to useful thermal energy. They also produce less nitrogen oxides.

Pulse combustors have been around for quite a number of years. Older readers and those with an enthusiasm for World War II technology will be familiar with the German V-1 "buzz bomb". These were propelled by pulsed combustors.

In America and Japan the principle is used for water and space heaters and in forced air furnaces. Their use in the UK has been severely limited because they are noisy—their harsh drone is a never-to-be-forgetten noise for those who lived in Southern England during the war—and they have a tendency to fail on the lightback test gases of British Standard 5258, Part 1.

The basic operation of the pulse combustor is very simple. The combustor consists of a combustion chamber with inlet flapper valves, a spark plug and a tuned tailpipe (see Figure).

Fuel and air are injected into the chamber through the flapper valves and the mixture is ignited by the spark plug, causing a pulse. The pulse closes the flapper valves and the combusting gases are forced out through the tailpipe. The tailpipe is tuned so that when the pressure wave reaches the end of the tailpipe it is reflected back to the combustion chamber. The partial vacuum created in the combustion chamber by the pressure wave going up the tailpipe causes the flapper valves to open and fuel and air enter the chamber. The returning pressure wave closes the valves and causes the mixture to ignite. The cycle starts all over again.

The spark plug is only needed to ignite the first cycle. After that the combustor is self-igniting and self-aspirating, but the reasons for this are not clearly understood.

In fact there is quite a lot that is not understood about pulse combustion, and research is going on in both the UK and USA to learn more about what happens in pulsed combustors and to make them more appropriate for the domestic market.

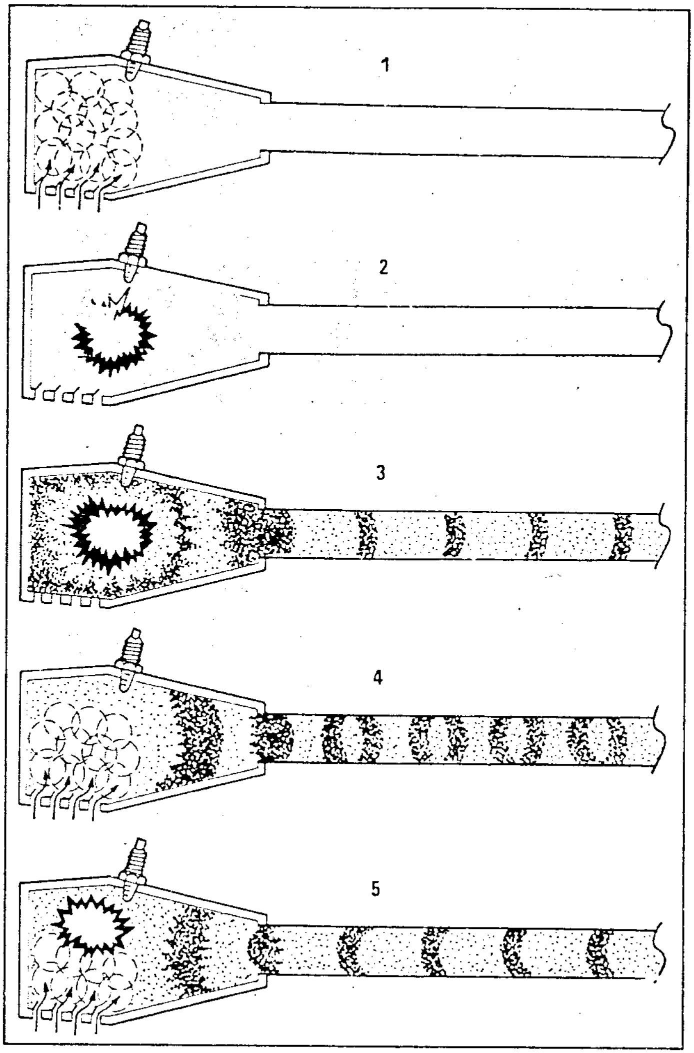
At Watson House in London, British Gas researchers are trying to solve the inherent noise and combustion problems associated with pulsed combustion. They are also trying to make full use of the powerful flue pressure in simplified heat exchanger design with remote, miniature flue applications. The basic goal is to produce a compact condensing boiler incorporated into a storage cylinder. Simple immersion tubes would provide the heat transfer surface within the cylinder and a miniature flue would be used to remove the combustion products.

Work is also being done on design guidelines to ensure satisfactory combustion on the British Standards test gases.

Further research is being sponsored at Leeds University and Essex University by British Gas. At Leeds researchers are looking at heat transfer aspects, and at Essex noise suppression is being examined.

In the USA at the Sandia National Laboratories facilities in Livermore, California, researchers are trying to get down to the basics of why they work, why they are more efficient and why they create less pollution. With the results they hope to produce scaling guidelines so that combustors of maximum efficiency can be produced at various sizes. Researchers also believe the work could be incorporated into other applications, such as wood or coal combustors, chemical reactors, coal gasification, gas turbines, electric generators and high temperature heat pumps and heat exchangers.

Sandia's research is in cooperation with Toshiba Corporation of Japan, and work is based on a Toshiba-designed water heater which has been adapted with quartz windows so that high-speed schlieren photography and laser Doppler velocimetry can be used.



The basic operation of a pulse combustor: (1) Fuel and air enter the combustion chamber through the flapper valves and (2) are ignited by the spark plug causing a pulse. (3) The pulse closes the flapper valves and forces the combusting gases down the tailpipe. (4) The partial vacuum created in the combustion chamber opens the flapper valves and draws in fuel and air. However, a pressure wave has been reflected back down the tailpipe, and when it re-enters the chamber (5) the new fuel/air mixture ignites without use of the spark plug.

The research combustor has a mixing chamber with counter flowing air and methane, which helps mixing, and a multi-section tailpipe. Varying the length of the tailpipe by adding or subtracting sections means that the frequency of combustion can be varied.

The schlieren photography gives information about the movement of combusting reactants. With the laser velocimetry 0.05 micron-diameter particles of alumina are mixed with the air before it enters the combustion chamber. These particles readily scatter light, and by measuring their velocity air movement in the chamber can be determined.

Dr Jay Keller, of the Combustion Applications Division at Livermore, says that research so far has raised more questions than it has answered. They are, however, beginning to understand why the pulsed combustors are efficient and clean burning.

Work was done on pulsed combustors in the 1940s and early 1950s when researchers believed that the pressure boost could be used as a means of propulsion possibly for aircraft and rockets. Interest waned as jet engine technology progressed. Interest returned when fossil fuels began their rapid price rise in the 1970s. Some researchers believe that they have a big future ahead of them, burning not only natural gas but a variety of other fuels.

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Pulsating Combustion

Using pulsating rather than steady combustors may improve the performance of systems and processes controlled by momentum, heat, and mass transfer

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Increasing numbers of highly efficient space and water heaters that employ pulse combustors have appeared on the market in recent years. These combustors have also been used successfully in industrial drying and are being considered for other industrial applications where pulsations may improve efficiency. Fuel savings, reduced pollutants, and increased combustion intensity, convective heat transfer, and mass transfer rates are among the features of pulsating combustors that may reduce the operating and capital investment costs of many systems and processes.

Pulsating combustion refers to a combustion process that varies periodically. In the internal combustion engine, the timing of the periodic combustion process is controlled by the electric discharges of the spark plugs. In pulsating combustors, on the other hand, the process occurs spontaneously after ignition. The "singing flame" phenomenon is probably the first known example of pulsating combustion. It was discovered in 1777 when a gas flame placed in a round, vertical tube caused the spontaneous excitation of one of the natural acoustic modes of the tube [1]. The resulting oscillation was controlled by the location of the flame within the tube, the dimensions of the tube, and the characteristics of the gas feed tube. Subsequent studies showed that flames are also sensitive to excitation from sound and that their response depends upon the amplitude and frequency of the sound wave and the part of the flame on which the sound impinges. This sensitivity is demonstrated by the movement of a candle flame in a room where music is playing.

The first attempts to use pulsating combustors were apparently made around the turn of the century when the incorporation of these devices in air breathing propulsion systems and gas turbines was investigated [2,3]. These pioneering efforts were probably inspired by the internal combustion engine, in which the repetitive combustion process is, in fact, a forced, pulsating combustion process.

The influence of the internal combustion engine was particularly apparent in the jet engine proposed by Lorin [2] in 1908. Here, combustion occurred periodically inside a cylinder with a reciprocating piston, and thrust was produced as the combustion products left the engine through the exhaust nozzle. The engine's piston was attached to a movable shaft that provided only enough power to compress the cylinder's contents prior to combustion. Another propulsion scheme, proposed by Marconnet in 1908 [2,4], was similar in its operating principles to the well-known pulse-jet engine developed by Schmidt in the 1930s (see Figure 1). This was the predecessor of the German V-1 rocket motor, or *buzz bomb* [2,4,5], which was used in the bombing of London in World War II. Karavodine and Holzwarth [3,4] attempted to use the exhaust flows of pulsating combustors to drive turbine wheels.

In 1933, F. H. Reynst filed his first patent application for the *combustion pot* [4], which burned coal under pulsating conditions. Reynst's entire scientific career, which continued through the mid 1950s, was devoted to the study of pulsating combustion. His collected works discuss a variety of pulsating combustion applications, including Schmidt's pulse jet, gas turbines, boilers, aircraft propulsion, and pulverized coal combustion. Many of these problems are of much interest today.

The literature of the past thirty years contains scattered papers on pulsating combustion. They include pulse jet studies in the United States and France [5], pulverized coal combustion for half-scale boiler applications in Germany [6], pulverized coal and liquid-fuel combustors in the Soviet Union [7,8,9], and more recent developments in gas burning pulsating heaters [4,10–15].

CHARACTERISTICS

Studies indicate that the pulsations within a pulsating combustor and in its exhaust flow enhance momentum, heat and mass transfer, and the processes that generally control the effectiveness of energy producing systems. For example, improved mixing of fuel and air within the combustor reduces the combustor size and/or the amount of excess air required to achieve complete combustion. The latter increases thermal efficiency by lowering exhaust flow losses which, in turn, reduces operating costs. Similarly, improved convective heat transfer reduces the required heat transfer area and associated capital costs. A disadvantage of pulsating combustors is that their noise must be reduced to acceptable levels. System components must also be capable of withstanding the mechanical stresses caused by the pulsations.

Pulsating combustion is achieved when the heat released by the combustion process spontaneously excites pressure waves (i.e., one of the system's natural acoustic modes) within the system. The interaction of the combustion process with the flow oscillations results in a periodic heat release rate. To maintain the oscillations, the energy supplied to the pulsations by the combustion process must compensate for the energy lost by the pulsations due to viscous dissipation, heat transfer, and radiation of acoustic energy out of the system. In order to drive the pulsations within the combustor, the periodic heat re-

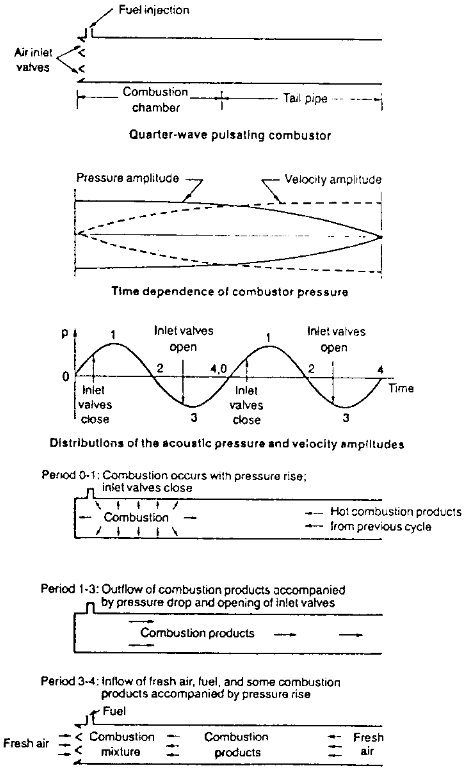
Fuel injection
Air inlet valves
Combustion chamber
Tail pipe
Quarter-wave pulsating combustor
Pressure amplitude
Velocity amplitude
Time dependence of combustor pressure
p
1
Inlet valves open
2
4,0
1
Inlet valves open
2
4
0
Inlet valves close
3
Inlet valves close
3
Time
Distributions of the acoustic pressure and velocity amplitudes
Period 0-1: Combustion occurs with pressure rise; inlet valves close
Combustion
Hot combustion products from previous cycle
Period 1-3: Outflow of combustion products accompanied by pressure drop and opening of inlet valves
Combustion products
Period 3-4: Inflow of fresh air, fuel, and some combustion products accompanied by pressure rise
Fuel
Fresh air
Combustion mixture
Combustion products
Fresh air

Figure 1. Characteristics of the quarter-wave pulsating combustor.

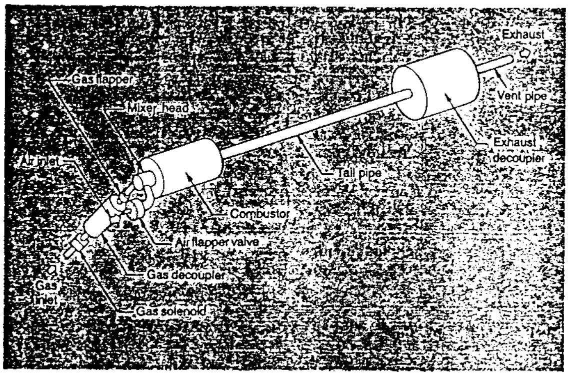

Figure 2. Helmholtz pulsating combustor.

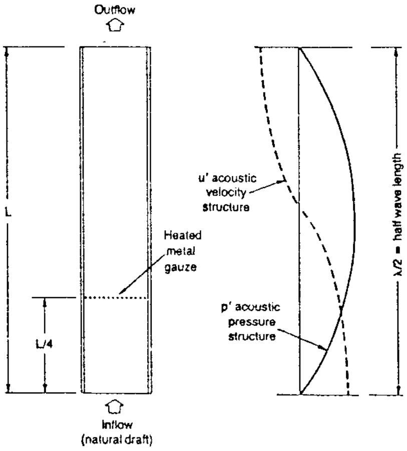

Figure 3. Rijke tube.

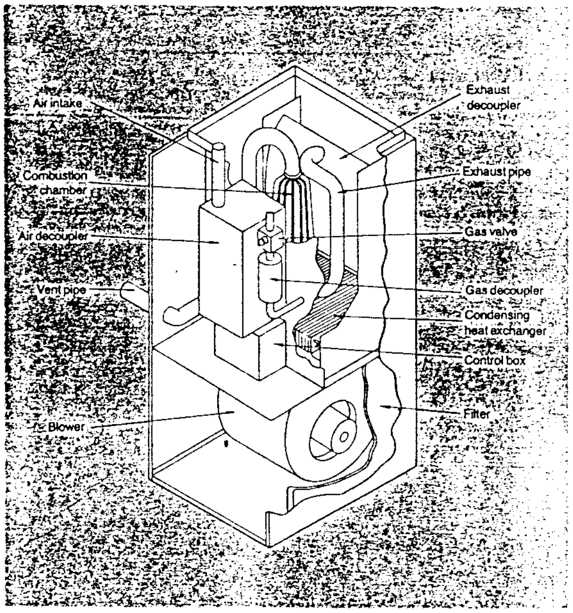

Figure 4. Pulsating combustion furnace.

lease must be in phase with the combustor pressure oscillations. This condition is referred to as Rayleigh's criterion. Finally, the combustion process heat release must occur within a region where the magnitude of the pressure oscillations is significant.

In principle, pulsating combustion can take place in any system that satisfies these conditions. The pulsating combustors that have been developed to date can be divided into three classes according to the acoustic system on which their operation depends. These are the quarter-wave tube, the Helmholtz resonator, and the Rijke tube.

Quarter-wave Combustor. This combustor is based on the operating principle of the acoustic quarter-wave tube, which consists of a pipe closed at one end and open at the other. The acoustic pressure oscillations excited within this tube have their maxima at the closed end and their minima (which are approximately zero) at the open end, as shown in Figure 1.

For purposes of description, the quarter-wave combustor can be divided into three subsections: the inlet, the combustor, and the tail or exhaust pipe. The inlet section consists of flapper valves that open when the combustor pressure becomes smaller, and close when it becomes larger than the pressure upstream of the valves. In contrast, the inlet section of the Marconnet engine, which appears to operate like a quarter-wave combustor, consists of an aerodynamic valve (frequently a short, open tube whose diameter is smaller than the combustor diameter) that offers high resistance to the flow of gases out of the combustor and low resistance to the flow of gases into it. The combustor section consists of the portion of the tube near the flapper valves where most of the combustion occurs. The exhaust pipe comprises the remainder of the tube.

A combustible mixture is first admitted into the combustor, where it is ignited by means of a spark plug. The ensuing combustion process increases the combustor pressure which, in turn, causes the flapper valves to close and the hot combustion products to move towards the exhaust pipe. The inertia of the gases leaving the combustor decreases the combustor pressure to a level lower than the pressure upstream of the flapper valves. Consequently, the flapper valves open and admit new charges of fuel and oxidizer into the combustor. The decrease in combustor pressure also reverses the flow in the tail pipe and returns some hot combustion products to the combustor. The new charges of fuel and oxidizer mix, react, and release energy, causing a new pressure increase in the combustor. This periodic combustion process can now repeat itself indefinitely without a spark plug.

The combustion process in the quarter-wave combustor occurs in the vicinity of the flapper valves where the magnitude of the pressure oscillation is close to its maximum. Furthermore, the fresh charges of combustibles are admitted into the combustor when its pressure is close to or at a minimum. Consequently, to satisfy Rayleigh's criterion, the duration of the mixing and reaction processes must be such that most of the combustion heat is released when the combustor pressure is near its maximum. Referring to the combustor pressure time-dependence diagram in Figure 1, this means that the mixing and reaction processes must be completed within, approximately, half a period of the pulsations. If, however, the duration of these processes and of the pulsations is too short to satisfy Rayleigh's criterion, the combustor will not operate.

Interestingly, the German buzz bomb probably operated as a quarter-wave combustor. The thrust was provided by the reaction force on the combustor walls resulting from the expulsion of the combustion products through the exhaust end. When the flapper valves are replaced by an aerodynamic valve, some of the combustion products escape through this valve when the combustor pressure is higher than the outside pressure. This outflow can be avoided by using fans to force the flow of fuel and oxidizer through the system.

Helmholtz Combustor. The Helmholtz combustor is distinguished from a quarter-wave combustor by its larger volume (see Figure 2). The acoustic Helmholtz resonator [16], on which its design is based, consists of an enclosure having a very short neck attached to a small-diameter hole in one of its walls. The tail pipe of the Helmholtz combustor is quite long and the combustor is attached to fuel and air inlet valves. In spite of geometrical differences, the Helmholtz combustor resembles the quarter-wave combustor in its operation.

Rijke Combustor. The third class of pulsating combustors is based on the operating principles of the Rijke tube [17]. (see Figure 3). A heated metal gauze placed at a distance (*L*/4) from the bottom of an open-ended, vertical tube (of length *L*) results in the excitation of the fundamental acoustic mode of the tube. An examination of Figure 3 shows that the length of the tube equals half the wavelength of the oscillation. Further, the acoustic pressure reaches its maximum at the center of the tube and is nearly zero at both ends. Both acoustic pressure and velocity are nonzero at the location of the heater. It is important to note that the heat addition process establishes a steady upward flow in the tube, which is essential to the excitation of the oscillations.

In Rijke pulsating combustors, the heated gauze is replaced by a combustion process that provides the energy for driving combustor pressure oscillations [18-21]. As in all pulsating combustors, the interaction of the flow oscillations with the combustion process establishes a feedback loop—the combustion process produces flow oscillations which, in turn, produce a periodic and highly intense combustion process.

APPLICATIONS

Pulsating combustors, in particular the gas burning, Helmholz type, are most commonly used in space and water heaters. The water and/or air are heated by the walls of the combustor, by the tail pipe, and by one or more additional heat exchangers that are often positioned downstream of the tail pipe. Lennox, Inc., the leading manufacturer of space heating furnaces in the United States, offers units with input capacities from 40,000 to 130,000 Btu/hr [10] (see Figure 4). Hydrotherm, Inc., another U.S. firm, manufactures water heaters with capacities from 50,000 to 150,000 Btu/hr [11] (see Figure 5). In Japan, Toshiba is marketing water heaters with input capacities in the range of 21,000 to 60,000 Btu/hr. Other companies, such as Tokyo Gas and Osaka Gas, are also developing pulsating combustors for various domestic and industrial applications.

These pulsating combustors are self-aspirating and do not require additional chimneys or fans to move the air, fuel, and combustion products through the system. Consequently, most of the energy produced during combustion can be recovered by cooling the combustion products to near room temperature prior to exhaustion out of the system. This results in high thermal efficiencies (on the order of 95 percent) and considerable fuel savings. The cooling process, however, also produces some corrosive condensates in the exhaust flow, which require that noncorrosive materials be used in the final heat exchanger stage. Other benefits include higher rates of convective heat transfer from the combustion products to the walls, due to the pulsations in the flow, and reductions in the formation of NO_x (generally in the 25 to 50 ppm range).

Various coal and liquid-fuel burning pulsating combustors have also been investigated [7-9, 18-25]. Promising results have been obtained with coal-burning Rijke combustors. Unpulverized coal particles are burned in a bed-like combustion zone supported by a metal grid near the center of the lower half of the combustor (see Figure 6). The coal is fed to the bed from above by an auger and the combustion air is supplied from below. Pulsations occur spontaneously after ignition and the air-fuel ratio can be controlled by varying the coal and air supply rates.

Tests have demonstrated that the Rijke combustor can burn unpulverized bituminous and subbituminous coals over wide ranges of coal feed rates and air-fuel ratios, including highly fuel-rich conditions. The latter suggests that this combustor could be used as a coal gasifier. Tests with a subbituminous coal indicate that the Rijke combustor can achieve high combustion efficiencies with little excess air. For example, in tests at the Georgia Institute of Technology [22], combustion efficiencies on the order of 92 to 95 percent were attained with stoichiometric coal/air mixtures and various coal feed rates. The combustion efficiencies increased to 95 to 97.5 percent with five percent excess air and to 97 to 99 percent with ten percent excess air. These efficiencies were attained in spite of the fact that combustion was completed within a distance of approximately 12 inches above the bed support and that the combustion zone temperature was relatively low due to high heat losses through the uninsulated combustor walls. This performance can probably be further improved by insulating the combustor walls. In contrast, stokers that also burn unpulverized coals require 30 to 60 percent excess air to achieve comparable combustion efficiencies [26].

Such high combustion efficiencies combined with small quantities of excess air result in high thermal efficiencies by reducing exhaust flow losses and pumping costs. Reduced excess air also increases the temperature of the combustion products and associated heat transfer rates. Finally, it should be pointed out that in the Soviet Union, experiments with pulsating combustion using liquid fuels [9] also attained high combustion efficiencies with low excess air values.

Observations of the combustion zone in the Rijke combustor reveal that the burning coal particles move in a spout-like motion that resembles the motion of particles in a fluidized bed combustor. This fluidization is caused by the acoustic velocity, which is greater than that of the steady flow in the combustor and which reverses its direction twice in each cycle. Studies conducted to date strongly suggest that flow reversals and pulsations in the combustion zone enhance gas phase mixing and increase the rate at which oxidizer molecules and heat are diffused to the fuel surface. This, in turn, increases the fuel combustion rate. Reynst argued that when coal particles burn in an oscillatory flow, a time-dependent relative velocity is established between the particles and the adjacent flow, resulting in periodic stripping of the blankets of combustion products that accumulate around the coal particles during the combustion process [4]. This stripping decreases the gas phase resistance to the migration of oxygen molecules to the particle surface, enhancing its burn rate. The increase in the burn rate of individual coal particles due to flow pulsations has been confirmed experimentally [27], and it is well known that pulsating combustors have high combustion intensities.

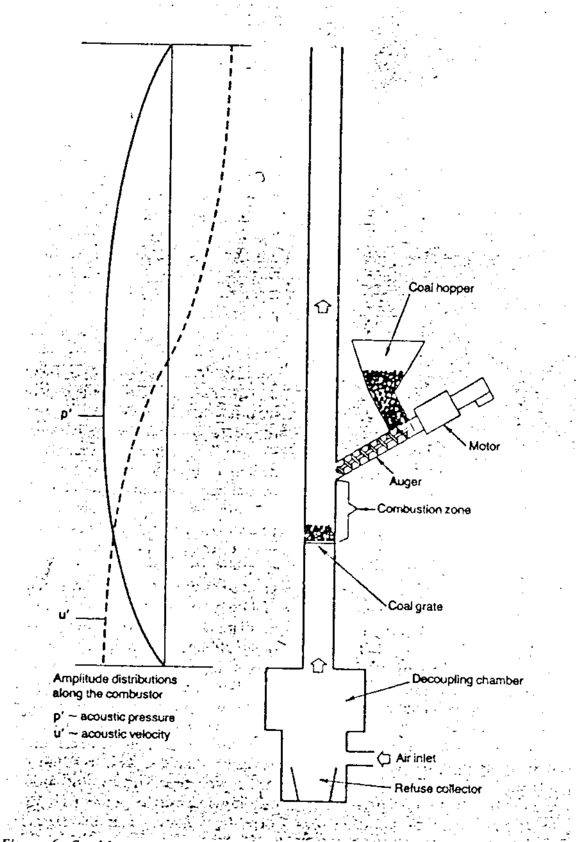

Figure 6. Coal-burning Rijke tube pulsating combustor developed at Georgia Tech and its acoustic wave structure.

The advantages of pulsating combustors suggest that they could be used effectively in various industrial processes. In recent years, two U.S. companies, Sonodyne and the Sonic Corp., have used pulsating combustors in drying processes. Slurries containing the material to be dried are injected into the pulsating exhaust flow of the combustors. The pulsations "shatter" the slurries and increase the rate of heat and moisture transfer to and from the dried substances. Other potential applications include coal gasification, pulsating fluidized beds, commercial deep-fat fryers, cooking equipment, radiant tube heaters, and chemical reactors.

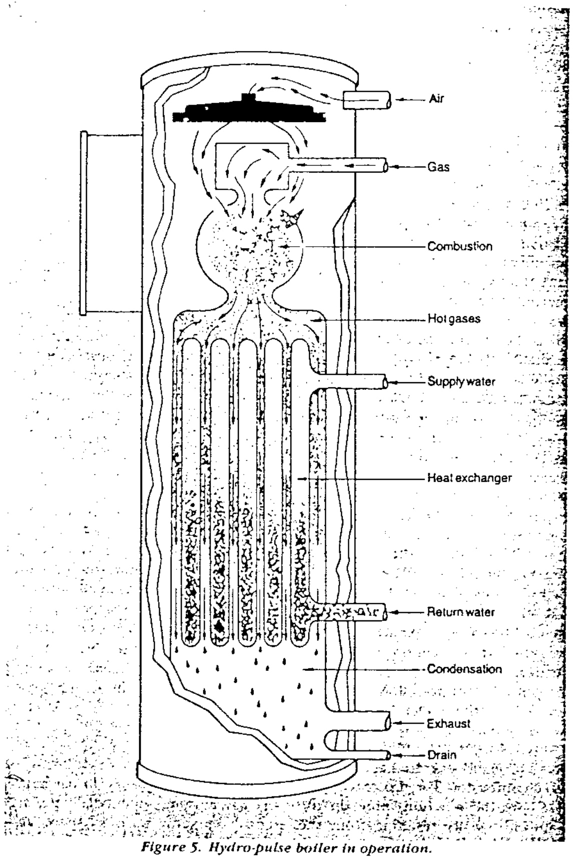

Figure 5. Hydro-pulse boiler in operation.

On the negative side, pulsating combustors are noisy and must be muffled to ensure an acceptable working environment. Sound treatment increases costs and may result in a bulky system when the sound is generated at low frequencies. Complications arise when several harmonics within the sound need to be suppressed. Finally, amplitudes increase as the energy supplied to the combustor increases. Consequently, there are incentives to develop pulsating combustors that operate at lower amplitudes and higher frequencies and generate noise consisting of a single frequency tone.

The successful application of pulsating combustors in heaters and dryers in recent years suggests that they can significantly improve the performance of many systems and processes whose behavior is controlled by momentum, heat, or mass transfer. However, much of the progress in this area has been achieved by costly trial-and-error approaches rather than by rational design based on an understanding of the periodic phenomena that occur within the systems. Consequently, unexpected difficulties may arise when attempts are made to change the scale and/or loading of these combustors or to use them in new applications that have different requirements. Such potential problems can be avoided by devoting more research to the fundamentals of pulsating combustion. ME

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A STUDY OF THE PULSATION DRIVING MECHANISM IN PULSATING COMBUSTORS

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SUMMARY. Experiments performed in a facility consisting of a Schmidt-type pulsating combustor, in which high speed photographs were taken and pressure, temperature and gas composition measured, showed that the air supply conditions at the inlet and the volume of the combustor, influence strongly the oscillation frequency. From the measurements, the existence of two separate regions, one containing cold air and the other fuel rich gas was found and a pressure-volume diagram drawn, showing the effect of chemical energy release and heat supply during the compression stroke and differentiating it from the expansion. A model of the interaction between the cyclic combustion process and the acoustic oscillations of the gas volume within the chamber and the tail-pipe is presented. The conditions for chemical energy release that result in high pressure amplitude are described.

INTRODUCTION

In pulsating combustors a proper feedback between the flow and the combustion process results in periodic variations of the flow properties inside the combustors.

The German Second World War V-1 rocket (better known as the buzz-bomb), that caused such damage and fear in London, is probably the best known example of a pulsating combustor. Although a great deal of work has since been performed on pulsating combustors [1-4], the general nature of unsteady combustion is understood only qualitatively, and there is not enough quantitative knowledge to predict the performance of an existing pulse combustion device or to design such a device to specification.

An important step towards achieving this goal is to gain a better insight into the mechanism that drives the pulsations. For this purpose an experimental facility, which includes a Schmidt-type pulsating combustor, equipped with pyrex windows for high speed photography, air and gas supplies and an appropriate ignition system was built. Measurements of pressure, temperature, gas composition and photography allowed a better insight to be obtained into the nature of the pulsation mechanism.

EXPERIMENTAL

The experimental facility, shown in Fig. 1, consists of a pulse combustion chamber, 1, which was built in two versions, a small one having a volume of $5 \cdot 10^{-4} m^3$ and a big one, with a volume of $1.1 \cdot 10^{-3} m^3$. The chamber has a variable length tail-pipe, 2, ranging from 0.31 to 0.91 m, and is provided with nine pairs of pyrex windows, 3, for photographic studies and an ignition system, 4. The experiments were performed with gas, stored in an external LPG tank, 6, while the air required in the ignition stage came from a compressor, 7.

The measurement system, indicated schematically by 8, included pressure transducers and a number of Pt/Pt-10%Rh thermocouples. The output of both was connected to a photorecorder. Gas composition was monitored by taking samples with a water-cooled probe, connected to a number of points. The samples were analyzed by conventional equipment. The highspeed camera, 5, Hycam 16mm, was operated at 2000 frames per second with Kodak sensitive color film.

RESULTS AND DISCUSSION

In a first series of experiments, in which the geometric dimensions of the inlet and outlet pipe and of the combustion chamber were varied, different resonating frequencies, ranging between 100 and 300 Hz were observed. It was found that the quarter length suggestion, put forward by Reynst[1] for this type of device, does not match well the actual results. Tail pipes of equal diameter but different length attached to the same combustion chamber caused a frequency change smaller that that predicted by the quarter length theory (see Fig. 2). Experiments in which stable operation of the combustor was achieved over a wide range of tail pipe lengths, up to a decrease by a factor of two, resulted in a frequency increase of less that 20%. One can see from the figure that air supply conditions at the inlet pipe, as well as changes of the chamber total volume influence strongly the oscillation frequency.

Typical pressure-time curves, gas air equivalence ratio and average temperature as function of time are presented in Fig. 3a. When gas is supplied by an inner nozzle directly to the combustion chamber, the fuel flow rate is constant for both the intake and exhaust periods; during backflow from the chamber a fuel-rich mixture is formed in the remaining gas volume. Endothermic reactions in the exhaust gas inside the chamber volume, namely

$$C_nH_m + n\ CO_2 \rightarrow 2n\ \ CO\ \ + \frac{m}{2}\ H_2 \qquad \text{and}$$

$$C_nH_m + n\ H_2O \rightarrow\ \ n\ CO + (\frac{m}{2} + n)\ H_2\ ,$$

are responsible for the temperature decrease, while an inverse flame formed there after the airjet enters the chamber, is responsible for temperature and pressure increases[5]. The average equivalence ratio over a cycle (measured directly) is 1.20; computations show that it goes down to 1.11 at the end of the suction stroke and reaches 1.27 at the end of the exhaust stroke. The corresponding gas temperature variations should be about 220°C; however the existence of two separate regions, one with cold air and the other with fuel rich gas during the suction period, reduces somewhat the actual average volume temperature.

The pressure-volume diagram, shown in Fig. 3b, explains the thermodynamic process within a pulsed combustor. Comparison with adiabatic compression and expansion, taking place in a non-reacting Helmholtz resonator, shows that in the combustion chamber chemical energy is released and heat is supplied during the compression stroke; this results in different curves for the pressure rise and drop processes and "drives" the oscillations.

In the high speed pictures air appeared as a fully transparent medium, the rich mixture, containing N_2, CO_2, CO, H_2O and H_2 appeared blue, while the high temperature combustion products were yellow to transparent in color.

From the fast photography data, together with gas composition and temperature measurements, the following picture emerged (see Fig. 4). The combustion cycle starts with an air inlet stroke; the temperature drops as the burned gases are exhausted and fresh air enters the combustor; the resonating effect and the temperature drop decrease the pressure in the chamber till it is mostly filled with fresh air and incomplete burning gas at a minimum temperature. The remnants of the hot combustion products reignite the fuel-air mixture and cause the flame to propagate, thus increasing the temperature and driving the pressure above atmospheric. This cuts off the fresh air supply and the exhaust stroke starts. The highest temperature is recorded as all the air in the chamber is used in the combustion process, thus filling the combustor mostly with completely burned products. Fuel continues to fill the chamber behind the escaping gases; the high temperature and the resonating effect cause the pressure to increase till it reaches a maximum in the overexpanded combustor. There the pressure begins to go down and as it falls below atmospheric, fresh air rushes in to mix with the fuel and the hot gases, starting a new cycle.

The cyclic combustion process interacts strongly with acoustic oscillations of the gas volume within the chamber and the tail pipe. When chemical energy of the burning process is released at a stage, having a positive derivative of the chamber pressure, both processes act in sequence and high pressure amplitude may be achieved. In our experiments on different shapes of the chamber, tail pipe and inlet, the maximal pressure amplitude was 60,000 Nm^{-2}.

These results show that there is considerable flexibility in designing pulse combustors and that the nature of the pulsations and their frequency are controlled by flame propagation within the chamber.

In future work simultaneous measurements of the instantaneous pressure in the chamber and at twelve points along the tail pipe, which would further improve the understanding of the governing mechanism, are planned.

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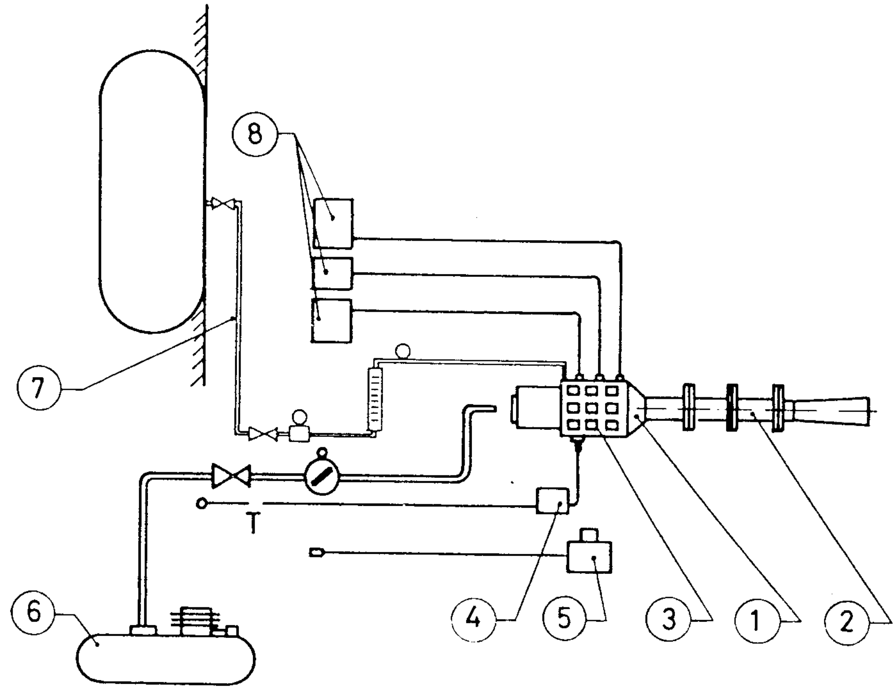

Fig. 1. Experimental facility.

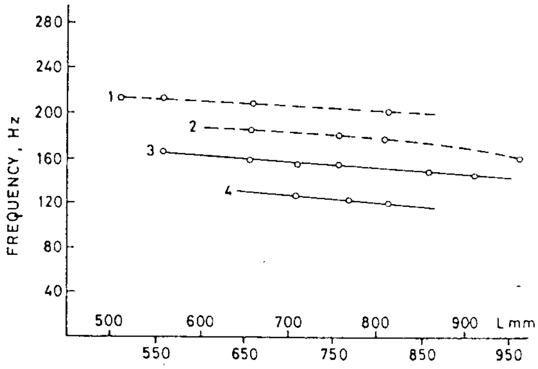

Fig. 2. Variation of frequency. 1 small chamber. 2 small chamber, air self supply. 3 big chamber, air self supply. 4 big chamber, pulsed air supply.

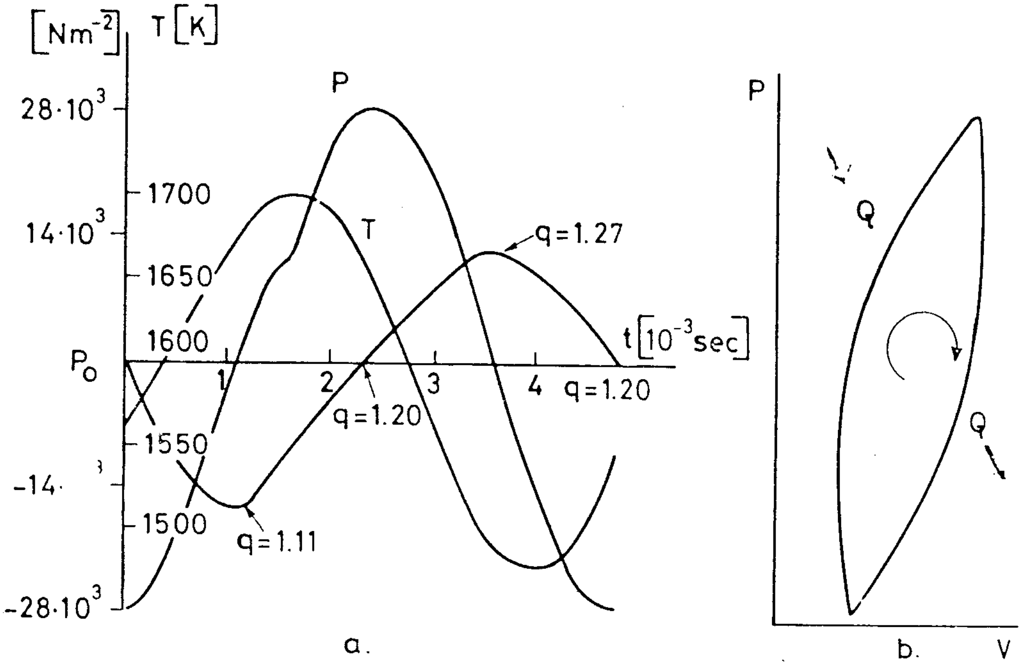

Fig. 3. a - Pressure, P; average temperature, T and equivalence ratio of gas in combustion chamber, q. b - P-V diagram of process.

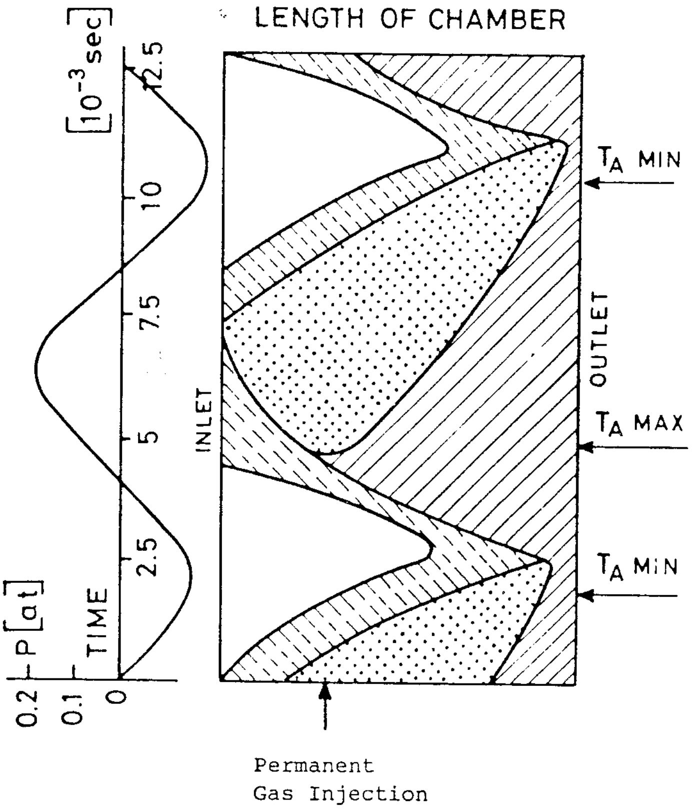

Fig. 4. Flame development in combustion chamber
☐ air ▒ rich mixture (blue)
▧ intense combustion ▨ complete combustion.

Stong

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4

THE "HILSCH" VORTEX TUBE

With nothing more than a few pieces of plumbing and a source of compressed air, you can build a remarkably simple device for attaining moderately low temperatures. It separates high-energy molecules from those of low energy. George O. Smith, an engineer of Rumson, N. J., discusses its theory and construction

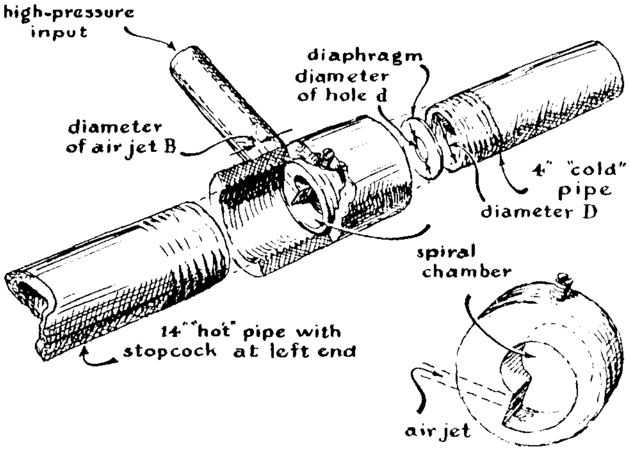
THE 19TH-CENTURY BRITISH PHYSICIST James Clerk Maxwell made many deep contributions to physics, and among the most significant was his law of random distribution. Considering the case of a closed box containing a gas, Maxwell started off by saying that the temperature of the gas was due to the motion of the individual gas molecules within the box. But since the box was standing still, it stood to reason that the summation of the velocity and direction of the individual gas molecules must come to zero.

In essence Maxwell's law of random distribution says that for every gas molecule headed east at 20 miles per hour, there must be another headed west at the same speed. Furthermore, if the heat of the gas indicates that the average velocity of the molecules is 20 miles per hour, the number of molecules moving slower than this speed must be equaled by the number of molecules moving faster.

After a serious analysis of the consequences of his law, Maxwell permitted himself a touch of humor. He suggested that there was a statistical probability that, at some time in the future, all the molecules in a box of gas or a glass of hot water might be moving in the same direction. This would cause the water to rise out of the glass. Next Maxwell suggested that a system of drawing both hot and cold water out of a single pipe might be devised if we could capture a small demon and train him to open and close a tiny valve. The demon would open the valve only when a fast molecule approached it, and close the valve against slow molecules. The water coming out of the valve would thus be hot. To produce a stream of cold water the demon would open the valve only for slow molecules.

Maxwell's demon would circumvent the law of thermodynamics which says in essence: "You can't get something for nothing." That is to say, one cannot separate cold water from hot without doing work. Thus when physicists heard that the Germans had developed a device which could achieve low temperatures by utilizing Maxwell's demon, they were intrigued, though obviously skeptical. One physicist investigated the matter at first hand for the U. S. Navy. He discovered that the device was most ingenious, though not quite as miraculous as had been rumored.

It consists of a T-shaped assembly of pipe joined by a novel fitting, as depicted in Figure 234. When compressed air is admitted to the "leg" of the T, hot air comes out of one arm of


high-pressure input
diaphragm diameter of hole d
diameter of air jet B
4" "cold" pipe diameter D
spiral chamber
14" "hot" pipe with stopcock at left end
air jet

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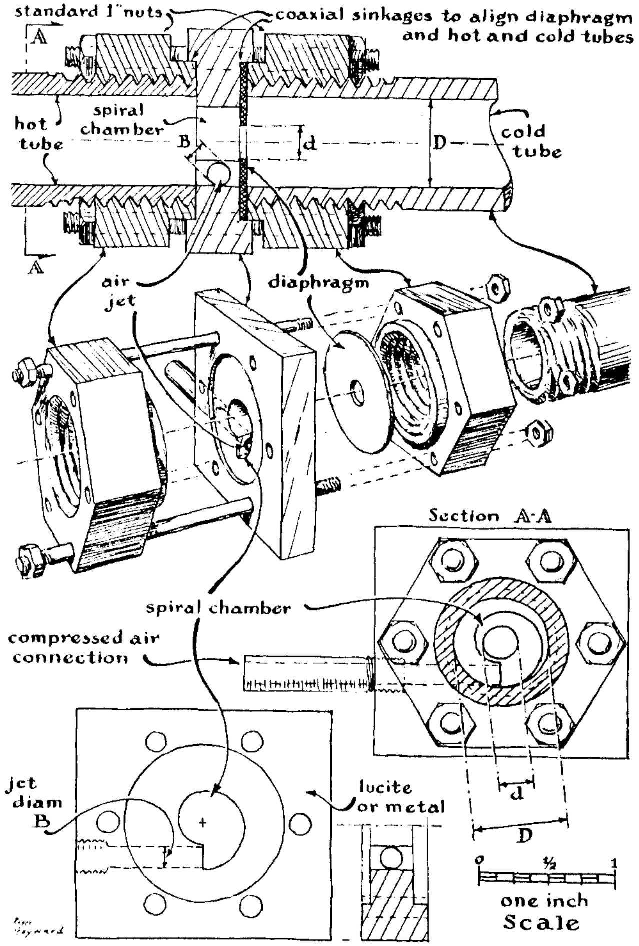
Cutaway view of Hilsch tube showing relation of spiral chamber to central pipe-coupling. Detail of spiral at lower right



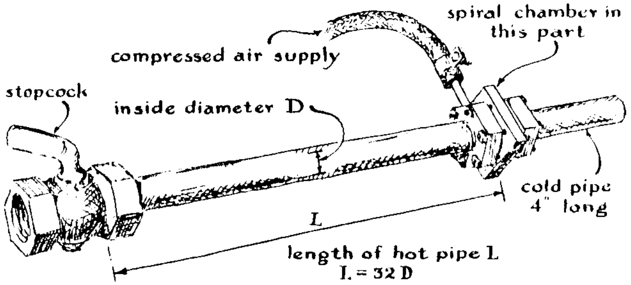
235
Version of Hilsch tube designed by Roger Hayward of Pasadena, Calif., for construction by amateurs. Note scale, lower right

the T and cold air out of the other arm! Obviously, however, work must be done to compress the air.

The origin of the device is obscure. The principle is said to have been discovered by a Frenchman who left some early experimental models in the path of the German Army when France was occupied. These were turned over to a German physicist named Rudolf Hilsch, who was working on low-temperature refrigerating devices for the German war effort. Hilsch made some improvements on the Frenchman's design, but found that it was no more efficient than conventional methods of refrigeration in achieving fairly low temperatures. Subsequently the device became known as the Hilsch tube.

The Hilsch tube may be constructed from a pair of modified nuts and associated parts as shown in Figure 235. The horizontal arm of the T-shaped fitting contains a specially machined piece, the outside of which fits inside the arm. The inside of the piece, however, has a cross section which is spiral with respect to the outside. In the "step" of the spiral is a small opening which is connected to the leg of the T.

Thus air admitted to the leg comes out of the opening and spins around the one-turn spiral. The "hot" pipe is about 14 inches long and has an inside diameter of half an inch. The far end of this pipe is fitted with a stopcock which can be used to control the pressure in the system [*see Fig. 236*]. The "cold" pipe is about four inches long and also has an inside diameter of half an inch. The end of the pipe which butts up against the spiral piece is fitted with a washer, the central hole of which is about a quarter of an inch in diameter. Washers with larger or smaller holes can also be inserted to adjust the system.



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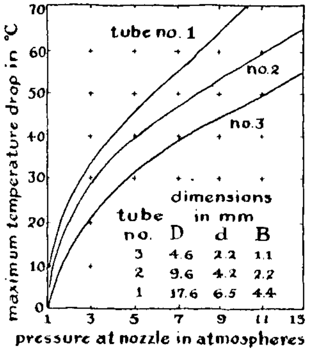
Relative lengths of hot and cold tubes of Hilsch device

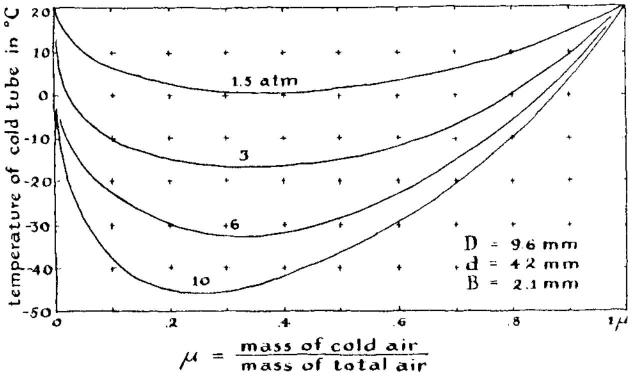
Three factors determine the performance of the Hilsch tube: the setting of the stopcock, the pressure at which air is admitted to the nozzle, and the size of the hole in the washer. For each value of air pressure and washer opening there is a setting of the stopcock which results in a maximum difference in the temperature of the hot and cold pipes [*see Fig.* 237]. When the device is properly adjusted, the hot pipe will deliver air at about 100 degrees Fahrenheit and the cold pipe air at about −70 degrees (a temperature substantially below the freezing point of mercury and approaching that of "dry ice"). When the tube is adjusted for maximum temperature on the hot side, air is delivered at about 350 degrees F. It must be mentioned, however, that few amateurs have succeeded in achieving these performance extremes. Most report minimums on the order of −10 degrees and maximums of about +140 on the first try.

Despite its impressive performance, the efficiency of the Hilsch tube leaves much to be desired. Indeed, there is still disagreement as to how it works. According to one explanation, the compressed air shoots around the spiral and forms a high-velocity vortex of air. Molecules of air at the outside of the vortex are slowed by friction with the wall of the spiral. Because these slow-moving molecules are subject to the rules of centrifugal force, they tend to fall toward the center of the vortex.

The fast-moving molecules just inside the outer layer of the vortex transfer some of their energy to this layer by bombarding some of its slow-moving molecules and speeding them up. The net result of this process is the accumulation of *slow-moving, low-energy* molecules in the center of the whirling mass, and of *high-energy, fast-moving* molecules around the outside. In the thermodynamics of gases the terms "high energy" and "high velocity" mean "high temperature." So the vortex consists of a core of cold air surrounded by a rim of hot air.



237
Graph of performance for three tubes made by Rudolf Hilsch. The tabulated dimensions "D" "d" and "B" refer to Figure 238



238
Performance graph of Hilsch tube showing effect of hot-tube stopcock adjustment for various input pressures

The difference between the temperature of the core and that of the rim is increased by a secondary effect which takes advantage of the fact that the temperature of a given quantity of gas at a given level of thermal energy is higher when the gas is confined in a small space than in a large one; accordingly when gas is allowed to expand, its temperature drops. In the case of the Hilsch tube the action of centrifugal force compresses the hot rim of gas into a compact mass which can escape only by flowing along the inner wall of the "hot" pipe in a compressed state, because its flow into the cold tube is blocked by the rim of the washer.

The amount of the compression is determined by the adjustment of the stopcock at the end of the hot pipe. In contrast, the relatively cold inner core of the vortex, which is also considerably above atmospheric pressure, flows through the hole in the washer and drops to still lower temperature as it expands to atmospheric pressure obtaining inside the cold pipe.

Apparently the inefficiency of the Hilsch tube as a refrigerating device has barred its commercial application. Nonetheless amateurs who would like to have a means of attaining relatively low temperatures, and who do not have access to a supply of dry ice, may find the tube useful. When properly made it will deliver a blast of air 20 times colder than air which has been chilled by permitting it simply to expand through a Venturi tube from a high-pressure source. Thus the Hilsch tube could be used to quick-freeze tissues for miscroscopy, or to chill photomultiplier tubes. But quite apart from the tube's potential application, what could be more fun than to trap Maxwell's demon and make him explain in detail how he manages to blow hot and cold at the same time?

Incidentally, this is not a project for the person who goes in for commercially made apparatus. So far as I can discover Hilsch tubes are not to be found on the market. You must make your own. Nor is it a project for the experimenter who makes a specialty of building apparatus from detailed specifications and drawings. The dimensions shown in the accompanying figures are only approximate. Certainly they are not optimum values. But if you enjoy exploration, the device poses many questions. What would be the effect, for example, of substituting a divergent nozzle for the straight one used by Hilsch? Why not create the vortex by impeller vanes, such as those employed in the stator of turbines? Would a spiral chamber in the shape of a torus improve the efficiency? What ratio should the diameter of the pipes bear to the vortex chamber and to each other? Why not make the spiral of plastic, or even plastic wood? One can also imagine a spiral bent of a strip of brass and soldered into a conventional pipe coupling. Doubtless other and far more clever alternatives will occur to the dyed-in-the-wool tinkerer.

5

A HOMEMADE INTERFEROMETER

Many of the material conveniences we take for granted depend on the optical effect that imparts color to soap bubbles and blueness to the bluebird. An application of the effect is the interferometer, most elegant of yardsticks. The principle is explained in the following pages and a design appropriate for amateur construction is suggested

When two rays of light from a single source fall out of step (say after they have taken different paths and met at a common point), their waves reinforce or counteract each other, just as out-of-phase waves in water do. This effect accounts for the blueness of the bluebird, the fire of opals, the iridescence of butterfly wings and the shifting colors of soap bubbles. It also accounts indirectly for the accuracy of watches, the control of guided missiles, the quality of high-test gasoline and myriads of other achievements of technology which would not be possible without precise standards of measurement.

All measurements, in the final analysis, depend upon one standard – length. Nowadays the length of the meter is calibrated in terms of wavelengths of light. The most commonly accepted standard for determining the length of the meter is the wavelength of the red light emitted by glowing cadmium in the vapor state. By interferometer methods this wavelength has been measured to high precision and comes out to be 6,438.4696 ± .0009 Angstrom units (an Angstrom unit is one 10-billionth of a meter). The meter is 1,553,164.60 ± .22 times this wavelength.

The interferometer, the most elegant of yardsticks, is a singularly finicky and frustrating gadget. A scientist once remarked: "Without a doubt the interferometer, particularly the

Newsgroups: sci.physics.fusion
From: bernecky@starbase.nl.nuwc.navy.mil (W. Robert Bernecky)
Subject: Wirbelrohr or vortex tube
Sender: scott@zorch.SF-Bay.ORG (Scott Hazen Mueller)
Date: Sat, 1 Jul 1995 23:11:02 GMT

The following may be relevant to the Potapov device.

It contains excerpts from "And yet it moves...strange systems & subtle questions in physics," by Mark P. Silverman, Cambridge University Press, 1993; Chpt 6 "The Wirbelrohr's roar".

"It was a Wirbelrohr, he explained; you blew into the stem, and out one end of the cross-tube flowed hot air, while cold air flowed out the other. I laughed; I was certain he was teasing me. Although I had never heard of a Wirbelrohr, I recognised a Maxwell demon when it was described."

"...he machined in his basement workshop a working model which I received from him shortly afterwards. The exterior was more or less just as he had described it: two identical long thin-walled tubes (the cross-bar of the T), were connected by cylindrical collars screwed into each end of a short section of pipe that formed the central chamber; a gas inlet nozzle (the stem of the T), shorter than the other two tubes but otherwise of identical construction, joined the midsection tangentially (Fig. 6.1). Externally, except for a throttling valve at the far end of one output tube to control air flow, the entire device manifested bilateral symmetry with respect to a plane through the nozzle perpendicular to the cross-tubes.

"Only someone with the lung capacity of Hercules could actually blow into the stem. Instead, the nozzle was meant to be attached to a source of compressed air. Taking the Wirbelrohr into my laboratory, I looked sceptically for a moment at its symmetrical shape before opening the valve by my work table that started the flow of room-temperature compressed air. Then, with frost forming on the outside surface of one tube, I yelped with pain and astonishment when, touching the other tube, I burned my fingers!"

"...With the few parts of the Wirbelrohr laid out on my table, I understood better the significance of the German name, Wirbelrohr, or vortex tube. The heart of the device is the central chamber with a spiral cavity and offset nozzle. Compressed gas entering this chamber streams around the walls of the cavity in a high-speed vortex. But what gives rise to spatially separated air currents at different temperatures? ...the placement in one cross-tube (the cold one) of a small-aperture diaphragm effectively blocked the efflux of gas along the walls of the tube, thereby forcing this part of the air flow to exit through the other arm whose cross-section was unconstrained.

```
                |-----------|
 --------------|           |-----------------
                           |   "COLD" PIPE
  "HOT" PIPE
                           | <--- diaphragm
 --------------|           |-----------------
                |---|   |---|
             /      |   |
    CENTRAL         |   |
    CHAMBER         |   |
         |           |   | <- INLET
       _____
      /      \                 Fig 6 - Schematic of Wirbelrohr or
    /   __     \               vortex tube.
   /   /        \
  |   /          |  Top View
  |   |          |
   \  | |       /
    \ | |      /
      | |    /
      | |---
      | |
      | | <- INLET
      | |
```

Room-temperature compressed air enters the inlet tube, spirals around the central chamber, and exits through the 'hot' pipe with unconstrained cross-section or through the 'cold' pipe whose aperture is restricted by a diaphragm.

"The glimmer of a potential mechanism dawned on me. Had the incoming air conserved angular momentum, the rotational frequency of air molecules nearest the axis of the central chamber would be higher - as would also be the corresponding rotational kinetic energy - than peripheral layers of air. However, internal friction between gas layers comprising the vortex would tend to establish a constant angular velocity throughout the cross-section of the chamber. In other words, each layer of gas within the vortex would exert a tangential force upon the next outer layer, thereby doing work upon it at the expense of its internal energy (while at the same time receiving kinetic energy from the preceding inner layer). Energy would consequently flow from the center radially outward to the walls generating a system with a low-pressure, cooled axial region and a high-pressure, heated circumferential region. Because of the diaphragm, the cooler axial air had to exit one tube (the cold side), whereas a mixture of axial and peripheral air exited the other (the hot side).

"The presence of the throttling valve on the hot side now made sense. If the low pressure of the air nearest the axis of the tube fell below atmospheric pressure, the cold air would not exit at all...By throttling the flow, pressure within the central chamber was increased sufficiently so that air could exit both tubes.

"...with some simplifying assumptions I was able to calculate the entropy change... Under what is termed adiabatic conditions - i.e. with no heat exchange with the environment - the 2nd Law requires that the entropy change of the gas, alone, be >= zero. The resulting mathematical expression, augmented by the equation of state of an ideal diatomic gas and the conservation of energy (1st Law) yields an inequality:

$$(x^{f})[(1-fx)/(1-f)]^{(1-f)} \geq (Pf/Pi)^{(2/7)}$$

where x= Tc/Ti
Tc is temperature of cold air
Ti is initial temperature
Pf is the final pressure
Pi is the initial pressure
f is the fraction of gas directed thru the cold side

"By setting the expression for the entropy change equal to zero, I could calculate the lowest temperature that the cold tube should be able to reach if the gas flow were an ideal reversible process. The result was astonishing. With an input pressure of 10 atmospheres and the throttling set for a fraction f= 0.3, compressed air at room temperature (20 C) could in principle be cooled to about -258 C, a mere 15 degrees above absolute zero! (The corresponding temperature of the hot side would have been 80 C.)

"...The first experimental demonstation of a vortex tube seems to have been reported in 1933 by a French engineer, Georges Ranque [1]. by German physicist Rudolph Hilsch came to the attention of American chemist R.M. Milton... In Hilsch's hands, proper selection of the air fraction f (~ .33) and an input pressure of a few atmospheres gave rise to an amazing output of 200 C at the hot end and -50 C at the cold end[2]. Hilsch, who was the one to coin the term Wirbelrohr, used the tube in place of an ammonia precooling apparatus in a machine to liquify air.

"...Milton was not satisfied with the interpretation of Hilsch and Ranque that frictional loss of kinetic energy produced the radial temperature distribution...."

M Kurosaka et al[3,4], in 1982, proposed a far different mechanism, supported by experiment.

"With a loud roar air rushes turbulently thru the Wirbelrohr, just as it does thru a jet engine or a vacuum cleaner. Buried within that roar, however, is a pure tone, a "vortex whistle" as it has been called...the vortex whistle can be produced by tangential introduction and swirling of gas in a stationary tube. It is this pure tone that is purportedly responsible for the spectacular separation of temperature in a vortex tube.

"The Ranque-Hilsch effect is a steady-state phenomenon - i.e. an effect that survives averaging over time. How can a high-pitch whistle - a sound that, depending on air velocity and cavity geometry, can be on the order of a few kilohertz - influence the steady component of flow? The answer...was by 'acoustic streaming'. As a result of a small nonlinear convection term in the fluid equation of motion, an acoustic wave can act back upon the steady flow and modify its properties substantially. In the absence of unsteady disturbances, the air flows in a 'free' vortex around the axis of the tube; the speed of the air is close to zero at the center (like a hurricane), increases to a maximum at mid-radius, and drops to a small value near the walls. Acoustic streaming, however, deforms the free vortex into a 'forced' vortex where the air speed increases linearly from the center to the periphery. Acoustic streaming and the production of a forece vortex, rather than mere static centrifugation, engender the Ranque-Hilsch effect.

"The experimental test could not be more direct. Remove the whistle, and only the whistle, and see whether the radial temperature distribution remains. To do this [Kurosaka] monitored the entire roar with a microphone and ...decomposed it into frequencies of which the discrete component of the lowest frequency and largest amplitude was identified as the vortex whistle. Next, he enclosed the Wirbelrohr inside a tunable acoustic suppressor: a cylindrical section of Teflon with radially drilled holes serving as acoustic cavities distributed uniformly around the circumference. Inside each hole was a small tuning rod that could be inserted until it touched the outer shell of the Wirbelrohr to close off

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the cavity, or withdrawn incrementally to make the cavity resonant at the specified frequency to be suppressed.

"To simplify the experimental test, he sealed off one output of the vortex tube and monitored with thermocouples the temperatuare difference between the center and periphery. In the absence of the suppressor, an increase in pressure produced, as I had noticed when experimenting with my own vortex tube, a louder roar and greater temperature difference. When, however, the acoustic cavity was adjusted to suppress only the frequency of the vortex whistle (leaving unaffected the rest of the turbulent noise), the temperature difference plunged precipitously at the instant the corresponding input air pressure was reached. In one such trial, the centerline temperature jumped 33 C, from -50 C to -17 C. With further increase in pressure, the frequency of the whistle rose, and as it exceeded the narrow band of the acoustic suppressor, the temperature difference increased again.

"Additional evidence came from a striking transformation in the natuare of the flow...Before the vortex whistle was suppressed, the exhaust air swirled rapidly near and outside the tube periphery in the manner expected for a forced vortex. Upon supprssion, however, the forced vortex was also abruptly suppressed; now quiescent at the periphery, the air rushed out close to the centerline."

"For all I know, the case of the mysterious Wirbelrohr is largely closed although, science being what it is, future version of that device may yet hold some suprises in store. I have sometimes wondered, for example, what would result from supplying a vortex tube, not with room-temperature air, but with a quantum fluid, like liquid helium, free of viscosity and friction.

The exorcism of the demon in the Wirbelrohr will not, I suspect, dampen one bit the ardour of those whose passion it is to challenge the 2nd Law. Despite the time and effort that has been frittered away in the past, others will undoubtedly try again. On the whole such schemes are bound to fail, but every so often, as in the case of Maxwell's own whimsical creation, this failure has its positive side: when, from the clash between human ingenuity and the laws of nature, there emerge sounder knowledge and deeper understanding."

[1] G. Ranque, "Experiences sur la Detente Giratore avec Productions Simultanees d'un Echappement d'air Chaud et d'un Echappement d'air Froid", J. de Physique et Radium 4(7)(1933) 112 S.

[2] R. Hilsch, "The Use of the Expansion of Gases in a Centrifugal Field as Cooling Process", Rev. Sci. Instrum. 18 (1947) 108.

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Eaton/Cutler-Hammer Corp. Industrial Optoelectronics
720 80th St. SW voice: 1-800-426-9184
Everett, WA 98203-6299 fax: 1-206-347-0544

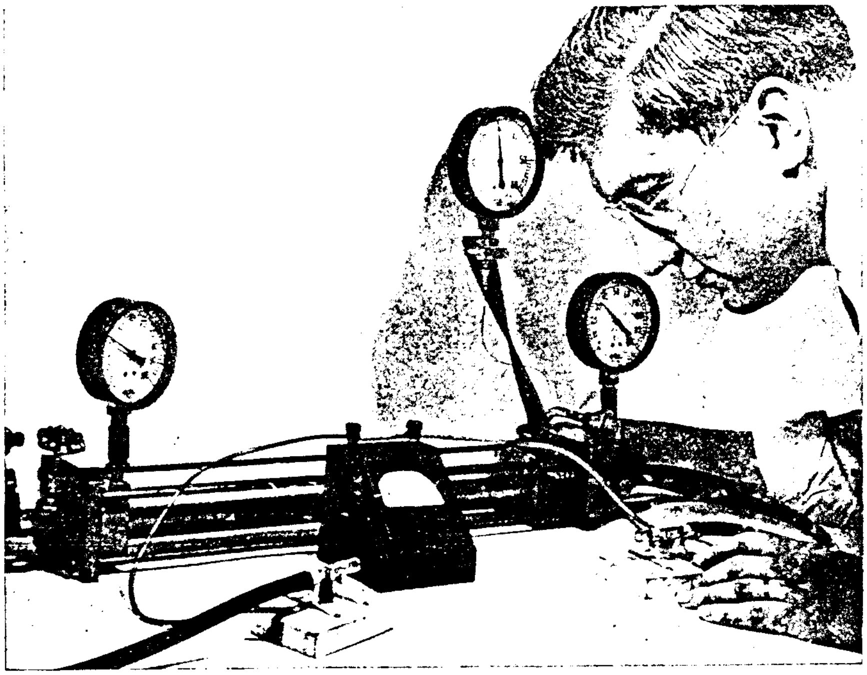
HOT AND COLD RUNNING AIR

A laboratory tool for making cold as simply as a Bunsen burner makes heat is being developed by Westinghouse research engineers. It is a pipe 15 inches long and one inch in diameter into which compressed air is pumped which is converted into a hot and a cold stream of air.

Within the pipe the air strikes the inner wall tangential to the diameter and is converted into a whirlpool of spinning gases—the air in the center becoming cold almost instantly while that on the periphery gets warm. The warm air can be tapped, leaving the jet of cold air to be drawn off through a ¼-inch opening at the center of the other end of the pipe.

As yet scientists have no explanation for the phenomenon, though it is conjectured that the cause may be a frictional effect between gases moving at different velocities. Also, its future applications in industry are not now foreseeable though even now it holds promise as a handy laboratory tool. Low efficiency rules out commercial applications at this time.

In the accompanying photograph, the right hand of the engineer is on the incoming compressed air line, while warm air is being drawn off at the left end of the pipe and cold air is coming from the opposite end.



Refrigerating Engineering, June 1947

How about capturing high velocity relative wind due to the car's motion, putting it into a vortex nozzle, and using the hot/cold plus heat pipes to heat or air condition the passenger compartment, and to heat the engine and cool the compressor simultaneously? This could neutralize part of the wind resistance by increasing the efficiency and decreasing the amount of fuel needed. The vortex tube added to a Tesla/Hodges/Neal/Rangely self-fueling pneumatic power plant would give us the first unlubricated, non-combustion, self-cooling, solar, permanent 99% efficient air-spring-powered, self-fueling automobile power plant with regenerative wind scoops.

Refrigerating Engineering
May 1950, p. 473-479

Journal of the ASRE

RANQUE'S TUBE

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Department of Mechanical Engineering
Massachusetts Institute of Technology

Since its discovery in Germany in 1945 by a Wartime Investigation Team, a vortex tube, which emits hot air from one end and cold air from the other when air under pressure is introduced tangentially near the middle of the tube, has been the source of speculation by refrigerating engineers concerning its practical applications and the cause of considerable controversy among heat transfer authorities in an effort to explain logically and mathematically the nature of the phenomena. In the February 1950 issue of REFRIGERATING ENGINEERING the article "An Analysis of the Hilsch Vortex Tube" was published. Some scientists and engineers disagreed with the theory of the tube's operation advanced in this article. In the accompanying article another theory is presented and a mathematical analysis is included in the Appendix. Readers are invited to comment on this analysis.

WHEN Georges Joseph Ranque stood before his colleagues of the Société Francaise de Physique in June 1933 and said that hot and cold air came out of a simple piece of pipe, he was received with skepticism.[1] He was a metallurgist associated with a steel company in the mountain town of Montlucon in central France; high-speed gas flow was out of his line.* No, said E. Brun, the aerodynamicist, stagnation temperature had been confused with static temperature and adiabatic wall temperature; the two streams weren't really cold and hot.[2] There is no further mention of the subject in the Société's Journal through 1944.

Ranque does not tell how he made his invention but it is possible he was concerned at some time with cyclone separators used to separate dust from air in steel plants. Air drawn from the center of such cyclones is slightly cool while air drawn from the outside is slightly warm. What would now be called the Ranque effect had been noticed, but apparently ignored, since such separators were first made.**

On December 12, 1931, Ranque filed a French patent docket on what is now known as the Ranque Tube, Hilsch Tube, Vortex Tube, or Vortex Refrigerator. After the French patent issued in 1932, he filed the same docket in the United States on December 6, 1932, and it issued on March 27, 1934.[3] He shows two basic types of design which may be called Counterflow and Uniflow, illustrated in Figures 1 and 2. He shows that the tangential entrance may consist of a single nozzle, a plurality of nozzles, or a set of blades. He describes how, by adjusting the size of the cold-air orifice or the restriction at the end of the hot tube, one may obtain a small quantity of very cold air or a larger quantity of moderately cold air. He mentions that the temperature of the hot tube reaches its maximum when the end of the hot tube is entirely closed, and that the more the pressure of the air supplied, the colder is the cold air. He speaks of having measured the pressure distribution inside the tube. Ranque must have come to hope he could accomplish significant things in refrigeration with his invention, for he assigns it to La Giration Des Fluides, or Whirl-Gas, of Montlucon, apparently a small company he had organized.

In attempting to broaden his claims, Ranque shows a model containing its own compressor, which he probably never tested. The compressor consists of a bladed rotor driven by an external motor. The air discharged from this rotor passes directly into stationary blades which guide the air tangentially into a uniflow tube. Not only would the rotor have to turn at an extreme speed but, as there is no cooling after compression, the cooler stream of air would still be warmer than the air supplied to the compressor.

The theory Ranque gives in the patent, later rejected, is as follows: The rotating gas spreads out in a thick sheet on the wall of the tube and the inner layers of this sheet press upon the outer layers by centrifugal force and compress them, thus heating them. At the same time the inner layers expand and grow cold. Friction between the layers is to be minimized, to which end the uniflow design is considered advantageous. The sheet is envisioned as having a rather sharp inner boundary, the center of the tube being filled with quiet gas.

The very different theory Ranque gave the Société will be taken up later. His remarks only amounted to 2½ pages in print, without illustrations or data, and he admits withholding further information for purposes of secrecy. The invention remained virtually unknown for more than 10 years. In view of the curiosity that demonstrations of the tube have invariably stirred up since 1946, it seems unlikely that Ranque made demonstrations. As to La Giration Des Fluides, it probably collapsed when it became clear that the device cannot compete with ordinary refrigeration machines. Ranque must have so lost interest by then that he was no longer inspired to publicize the invention. It is to be hoped that a full account of his work will be published.

In the spring of 1945, scientists sent to learn of wartime German developments found the tube being studied by Rudolf Hilsch, physicist, of the University of Erlangen. A working model was brought to the United States and this, combined with the publication of Hilsch's well known paper, started the present revival.[4] The writer learned of the tube on December 6, 1946, while associated with the General Electric Company, through A. B. Hubbard of that company. Interest in the tube now seems to be world-wide. It is a favorite subject for study in the schools and for demonstrations in the laboratory and shop, and engineers are still discovering it for the first time and temporarily entertaining hopes of revolutionizing the refrigeration industry, as did the writer in his turn.

Hilsch refers briefly to Ranque's paper of 1933 as the source of the idea, but it seems that he had not learned of the patent. He had arrived nevertheless at exactly the same design shown in certain of Ranque's drawings. Following Hilsch, nearly everyone has used similar designs to the complete neglect of the uniflow type so far as the writer has learned. Hilsch mentions a hope of using the device for large-scale refrigeration such as the cooling of mines, but he has since become aware of the low efficiency.* He has actually used it in the liquefaction of gases.

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* Deduced from publications of Ranque, 1934-1943.

** Private communication from A. J. Nerad.

Because Ranque's work has been virtually unknown, it has been widely asumed that Hilsch is the originator of the device, and the name "Hilsch Tube" has gained headway. It is hoped that this error will gradually be rectified, while Hilsch's contributions will still be appreciated.

Since 1946, writers of widely varying backgrounds have undertaken to explain the device, but with infrequent success. This may be attributed to the fact that while the device itself is extremely simple, the processes occurring in it are among the most difficult in gas dynamics. Few gas dynamicists giving attention to the problem have yet felt ready to publish their work. Nevertheless there is more sound theory in existence than is generally appreciated. Most notable is the work of Kassner and Knoernschild.[5] In *Part I* the writer presents a slightly different approach, less suited to prediction of the temperature change but intended to explain the mechanism simply. The work of K. & K. is briefly described.

The question of the possibility of commercial application is in need of clarification because no adequate analysis of the efficiency has appeared, and the enormous power consumption that would be required for most applications is not generally appreciated. This is dealt with in *Part II*.

PART I. THE MECHANISM

The present status of the theory, for both uniflow and counterflow types, may be summarized as follows: Fresh gas, before it has travelled very far in the tube, succeeds in forming an almost free vortex in which the angular velocity or rpm is low at the periphery and very high toward the center. But friction between the layers of gas undertakes to reduce all the gas to the same angular velocity, as in a solid body. This causes the inner layers to slow down and the outer layers to speed up as the gas moves along, amounting to the flow of work from the center to the outside of the vortex. At the same time, because the center of the vortex is much colder than the outside, heat flows toward the center, but not so rapidly as the work flows. The inner gas is originally cooled by its expansion; it stays partly cold by giving away its kinetic energy to the outer gas by friction without receiving as much heat energy in return. The outer gas in turn receives more kinetic energy than it loses heat energy, and this kinetic energy eventually becomes converted into internal energy through friction in the hot end of the tube. The flow is highly turbulent and the free vortex is supersonic.

The typical flow pattern in the counterflow type would be something like that shown in Figure 3. The solid lines show stream-surfaces of revolution about the axis. The position at which a particle enters determines its stream-surface, upon which it then describes a spiral path with ever-decreasing radius.** There is a null or stagnation point on the axis at *O*, and the stream-surface through this point divides what becomes the cold gas from what becomes the hot gas. The chief uncertainty about the flow pattern is the location of the null point, which will shift to the right and left as the flow of cold gas is decreased and increased, respectively. No data on this question are known to the writer but it is thought that Figure 3 may represent a reasonable estimate for equal hot and cold flows.

SYMBOLS

T=static temperature, deg F abs
p=pressure, any units
c_p=specific heat at constant pressure, Btu per lb, °F
c_v=specific heat at constant volume, Btu per lb, °F
k=c_p/c_v
J=778.26 ft-lb per Btu
R=gas constant, ft per °F abs
σ=ratio of weight-rate of flow of cold gas to total flow
η=efficiency
ln=natural logarithm
W=total rate of flow, lb per sec
a=total availability, ft-lb per lb
a_T=availability due to temperature, ft-lb per lb
a_p=availability due to pressure, ft-lb per lb
h=enthalpy, Btu per lb
s=entropy, Btu per lb, °F abs
ρ=density, slugs per ft^3=lb-sec^2 per ft^4
r=radius, ft
θ=angle, radians
z=axial distance, ft
u, v, w=velocity components in the r, θ, z directions, ft per sec
ϕ=velocity potential, ft^2 per sec
Ψ=stream function, lb-sec per ft

M=Mach number = velocity ÷ $\sqrt{kgRT}$
ω=v/r, angular velocity, rad per sec (not the rotation of fluid mechanics)
μ=absolute viscosity, lb-sec per ft^2
τ=shear stress, lb per ft^2
C=ωr^2, constant in a free vortex
P=rate of shear work, ft-lb per sec
q=rate of flow of heat, Btu per sec
g=32.16 ft per sec^2
S=surface area, ft^2
κ=thermal conductivity, Btu per sec, ft, °F
$N_{Prandtl}$=$\mu\ g\ c_p/\kappa$, Prandtl number, dimensionless

Subscripts

$_1$ state of gas supplied to Ranque tube
$_c$ state of cold gas leaving Ranque tube
$_b$ body being cooled at fixed temperature
$_2$ state at end of reversible adiabatic expansion from 1 to p_c
$_0$ dead state—environmental temperature and pressure
$_j$ nozzle jet

Superscript

* turbulent

At every point in what Ranque calls the "chamber of action," extending from the orifice plate to, say, the null point, the centrifugal work flow exceeds the centripetal heat flow, resulting in a net outward flow of energy, shown by radial arrows. For this centrifugal energy flux to result in a decrease in stagnation temperature of the inner gas, it must cross the stream-surfaces. The stream-surfaces near the orifice plate, at *a*, are mainly parallel to the radius, and therefore the gas in those stream-tubes only transfers energy within itself and emerges at the periphery of the discharge orifice scarcely cooled at all. This has been verified experimentally by bringing up a small tube to the mouth of the orifice and splitting the efflux into inner and outer parts. Under conditions where air was supplied at 105 psia and 70 F and ⅓ of the total flow issued through the cold-air orifice, the mixed stagnation temperature of the cold air was —30 F; when an outer annulus containing ¼ of the cold-air flow was separated, it had a stagnation temperature of 30 F while the remaining core had a stagnation temperature of —50 F. This diluting of the cold gas may be considered a disadvantage of the counterflow type since a separation at the orifice would always be necessary if one wishes to obtain the lowest possible temperature. In the uniflow type this would be unnecessary.

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* Private communication from Prof. Meissner, Munich, Sept. 1949.

** Actually, since the flow is turbulent, we can only speak of the *mean* paths.

Slightly downstream of the chamber of action, the gas has a graded stagnation temperature ranging from rather neutral at the axis to very high at the wall. That is why the tube wall in that region is hotter than the final mixed hot gas, and hotter than the tube wall either at the far end of the tube or at the inlet end of the tube.

Wall friction apparently has three beneficial, but small, effects. First, it removes some of the initial angular velocity of the outer layers, thus increasing slightly the initial value of the shearing friction. Second, it tends to keep the outer layers at a low angular velocity instead of permitting them to speed up as they receive energy from the inner layers, thus permitting the energy migration to proceed further. Third, it causes a small axial work flow down the hot tube by retarding the rotating mass downstream. This axial energy flux is indicated by the axial arrows in Figure 3.

The lowest possible temperature in the cold gas by the present theory would be that obtainable by a reversible and adiabatic expansion from the initial state to the final pressure:

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1} \right)^{(k-1)/k} \tag{1}$$

If the cold gas could give away all its kinetic energy without receiving any heat, it would emerge at this final temperature. So far about 1/2 of this temperature drop has been attained in the mixed cold gas, and about 4/7 in the core of the cold gas. Considering all the limitations of the actual process in the tube, it would appear that little further improvement could be expected. The process in the tube is highly irreversible.

Further details of the mechanism are given in the Appendix.

PART II. THE EFFICIENCY

The two common methods of heating—combustion and the passage of an electric current through a resistance—cannot be reversed. One places a higher value on refrigeration than on heating because refrigeration is so much harder to obtain. It is for this reason that in almost no case are we interested in using the hot gas from Ranque's tube. While it contains a good deal of thermodynamic availability due both to its temperature and to its pressure, there is usually no way to use these availabilities directly in the application, and to add auxiliary apparatus to recover them would so encumber the equipment that the original advantage of simplicity would be lost.

We must distinguish between two basic problems in refrigeration. On the one hand there is the problem of abstracting heat from a body always at a fixed temperature and pumping this heat out into the environment. The domestic refrigerator is an example of this problem. For this purpose a Carnot refrigerator is an ideal machine while a compression machine is a practical machine. A simple stream of cold gas is not suitable for this purpose; its use would involve irreversibility and inefficiency.

On the other hand there is the problem of cooling a substance down from environmental temperature to a certain low temperature. The cooling of air for air conditioning is an example of this problem. A proper machine for this purpose would be an isothermal compressor followed by an adiabatic expander. If a simple Carnot machine or a simple compression machine is used for this purpose, the efficiency of the process is only about ½.

If a machine which is ideal for one purpose is used as a criterion for a machine serving the other purpose, the criterion is not basic. We shall first look into some nonbasic criteria for Ranque's tube as a producer of cold gas, and then into a basic criterion.

Nonbasic Criteria

Carnot Criterion—We may define a coefficient of performance for Ranque's tube as equal to the heat taken out of the cold gas divided by the work of reversible isothermal compression:

$$\text{C.O.P.} = \sigma \frac{J\, c_p\, (T_1 - T_c)}{R T_1 \ln \dfrac{p_1}{p_c}} \tag{2}$$

where p_1 and T_1 are the pressure and temperature of the gas supplied to the tube, p_c and T_c are the pressure and temperature of the cold gas leaving the tube, c_p is the specific heat at constant pressure, R is the gas constant of the gas, J is the mechanical equivalent of heat, and σ is the ratio of the weight-rate of flow of cold gas to the total flow.

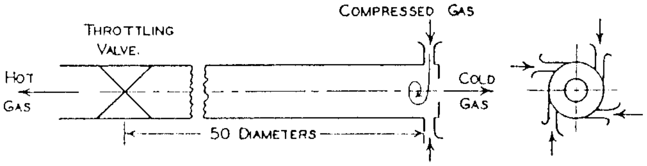


Fig. 1. The counterflow type.

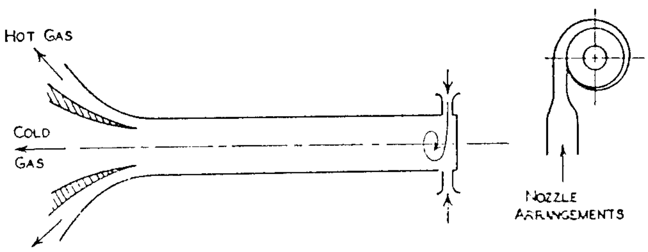


Fig. 2. The uniflow type.

The coefficient of performance of a Carnot refrigerator working between the temperatures T_c and T_1 is equal to $T_c / (T_1 - T_c)$. If we divide Equation (2) by this quantity, we obtain what may be called the efficiency of Ranque's tube on the Carnot basis:

$$\eta_{Carnot} = \sigma \frac{J\, c_p\, (T_1 - T_c)^2}{R T_1 T_c \ln \dfrac{p_1}{p_c}} = \sigma \frac{\left(1 - \dfrac{T_c}{T_1} \right)^2}{\dfrac{T_c}{T_1} \ln \dfrac{T_1}{T_2}} \tag{3}$$

In the second of these forms, T_2 is the temperature that would be reached in a reversible adiabatic expansion from p_1 and T_1 to p_c. We must be careful in interpreting the meaning of Equation (3).

Suppose we are interested in obtaining cold air at a temperature T_c. Then if, on the one hand, we use Ranque's tube to obtain that cold air and, on the other hand, we use a Carnot machine always working at T_c to obtain that cold air, Equation (3) gives the ratio of the works required by the two methods to produce the same amount of cold air, or the ratio of the amounts of cold air produced for the same work. Equation (3) is a meaningful criterion provided the possible competitor one wishes to consider can be no better than a machine of the fixed-temperature type, such as a Carnot machine or a conventional compression machine. Actually one would not be using the Carnot machine properly; it would be irreversibly and inefficiently applied. It turns out that an ideal machine would require only ½ the work of this inefficiently applied Carnot machine, for small temperature drops.

The efficiency of Equation (3) has been in the minds of several writers but for some reason it does not seem to have appeared in print. In Hilsch's small tube (Fig. 5) this efficiency reaches a maximum value of about 2.5 percent at 11 atmospheres pressure. For some reason there has been an impression that this efficiency is over 10 percent. The efficiency of Equation (3) is approximately twice that of the basic criterion shown in Figure 6, and so from that figure one can perceive how it varies.

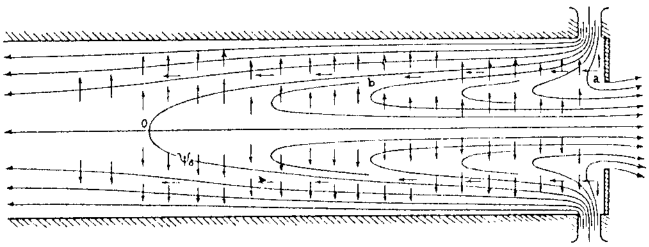


Fig. 3. Supposed flow pattern.

Criterion for Refrigeration at Fixed Temperature—Suppose we wish to use the cold air to remove heat from a body at a fixed temperature T_b. Then the cold air can only be utilized from T_c up to T_b, so that T_1 in the numerator of Equation (2) becomes T_b. Also T_c in the Carnot C.O.P. becomes T_b. Then the efficiency becomes:

$$\eta_{\text{Fixed temp}} = \sigma \frac{J\, c_p\, (T_b - T_c)\; (T_1 - T_b)}{R T_1 T_b \ln \dfrac{p_1}{p_c}} \tag{4}$$

This becomes zero if $T_b = T_c$ because then the cold air is not being utilized at all. It also approaches zero as T_b approaches T_1 if T_c does not also approach T_1, because then the work of the Carnot machine vanishes. It tends to be a maximum when T_b is halfway between T_c and T_1, and at that condition it has roughly ½ the value of the basic criterion shown in Figure 6, or a maximum of around 0.6 percent. The approximate factor of 4 between Equations (3) and (4) when T_b is halfway between T_c and T_1 is due to the fact that in Equation (4) the Carnot machine is only working half as hard while at the same time only half the refrigeration in the cold gas is being utilized. The Carnot machine is now being used properly while the cold gas is being used improperly, giving a deficiency of a factor of about 2 compared with the basic criterion.

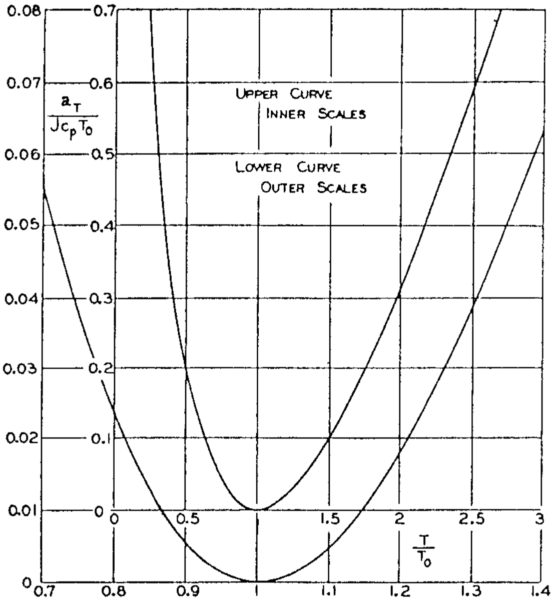


Fig. 4. Availability due to temperature of a perfect gas in steady flow.

Turbine Criterion—Suppose the competitor against which we wish to compare Ranque's tube is a reversible and adiabatic turbine producing the same amount of cold gas σ W at the same Temperature T_c, the work of the turbine being lost. This turbine will require a smaller pressure ratio than Ranque's tube, and also its compressor will only have to handle the flow σ W, instead of the flow W. The ratio of the work of the turbine's compressor to the Ranque tube's compressor turns out to be

$$\eta_{\text{Turbine}} = \sigma \frac{J\, c_p \ln \dfrac{T_1}{T_c}}{R \ln \dfrac{p_1}{p_c}} = \sigma \frac{\ln \dfrac{T_1}{T_c}}{\ln \dfrac{T_1}{T_2}} \tag{5}$$

where T_2 is the final temperature in a reversible adiabatic expansion from p_1 and T_1 to p_c.

This efficiency is rather insensitive to the pressure, unlike the efficiencies of Equations (3) and (4), and furthermore it is slightly higher at low pressures. In Hilsch's small tube (Fig. 5) at 11 atmospheres it has a maximum of about 13 percent at a σ of about 0.6, while at 2.5 atmospheres it has a maximum of about 15 percent at a σ of about 0.7. It is the loss of all the work from the turbine which permits Ranque's tube to show up much better in this criterion. Where a turbine is being used in this way, Ranque's tube is perhaps within a factor of 4 of competing, allowing for inefficiency in the turbine.

Hilsch's Criterion—It appears to the writer that what Hilsch calls the efficiency is the coefficient of performance of Equation (2). For the curves of Figure 5, at 11 atm this has a maximum of about 0.12 at a σ of about 0.6, at 7 atm and 4 atm a maximum of about 0.14 at a σ of about 0.65, and at 2.5 atm a maximum of about 0.13 at a σ of about 0.6. For a larger tube Hilsch makes calculations which seem to show values 40 percent to 60 percent higher.

The Basic Criterion

The basic criterion calls for comparing Ranque's tube against a reversible producer of cold gas, such as a reversible isothermal compressor followed by a reversible adiabatic expander. Let us make this comparison using the concept of availability.

The availability of a system is defined as the minimum amount of work required to bring the system from the dead state to the given state, operating in a large environment at a given temperature and pressure. It is also the maximum amount of work that can be obtained in reducing the system to the dead state. It is equal to the amount of work by any reversible process.

It is shown by Keenan[6] that the availability of a pure substance flowing steadily in the absence of changes in electricity, magnetism, capillarity, gravitational position, and velocity is given by

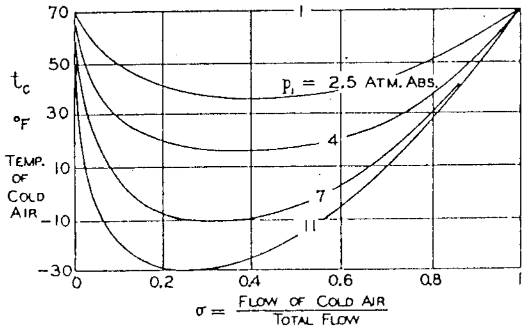
$$a = J\left[(h - h_o) - T_o\,(s - s_o)\right] \qquad (6)$$

in which the subscript, $_o$, refers to the dead state—viz., that state in which the substance has the pressure of the environment, p_o, and the temperature of the environment, T_o.

For a perfect gas, Equation (6) reduces to

$$a = Jc_pT_o\left[\frac{T}{T_o} - 1 - \ln\frac{T}{T_o} + \frac{k-1}{k}\ln\frac{p}{p_o}\right] \qquad (7)$$

In Equation (7), a has the units of ft-lb per lb. The last term represents the availability due to the pressure of the gas, and it will be recognized that this is simply the work of isothermal compression or expansion. If the pressure is lower than that of the environment, the last term is negative because work must be expended to pour the gas out into the environment.



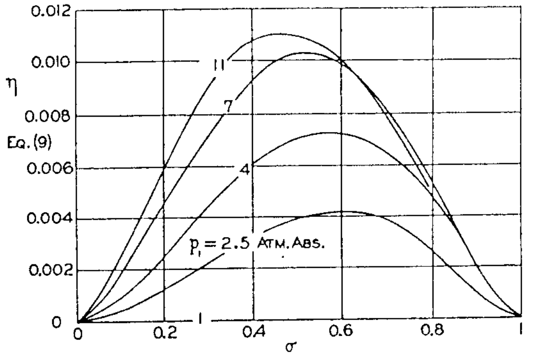


Fig. 5. (upper) Data of Hilsch for his small tube. Tube diameter 0.192 in., orifice diameter 0.0916 in., diameter of the single nozzle 0.0459 in., total flow at 11 atm 18.2 lb per hr or 4.12 cu ft per min requiring 0.623 hp for reversible isothermal compression. Cold air discharged at 1 atm abs.

Fig. 6. (lower) Basic efficiency as producer of cold air only from Figure 5 and Eq. (9).

The first three terms are a more interesting function; their sum represents the availability due to the temperature of the gas, and it is a property of this function that it has a positive value whether $T / T_o > 1$ or $T / T_o < 1$. That is because a heat engine can be operated using the gas either as a source of heat or as a sink, according to whether $T > T_o$ or $T < T_o$. The reversible process required can be accomplished by using an infinite series of Carnot engines, each operating at a slightly lower or higher temperature than the last, or much more simply by an adiabatic compression or expansion followed by an isothermal expansion or compression to restore the pressure. In Equation (7) the first two terms amount to $Jc_p\,(T - T_o)$; this is the heat exchanged with the gas. The third term is the amount of heat exchanged with the environment. The difference is the amount of work involved.

Figure 4 shows how the availability due to temperature varies. It is seen that for small temperature changes it is approximately proportional to the square of the temperature drop. Therefore a pound of gas at 10 deg F below the environment is worth approximately four times as much as a pound of gas at 5 deg F below the environment. In calculating the availability for small temperature drops, one is subtracting nearly equal numbers in Equation (7), and it is advisable to expand the logarithm in a series, giving

$$a_T = Jc_pT_o\left[\frac{1}{2}\left(\frac{T}{T_o} - 1\right)^2 - \frac{1}{3}\left(\frac{T}{T_o} - 1\right)^3 + \ldots\right] \qquad (8)$$

We now define the efficiency of the Ranque tube as a producer of cold gas as equal to the work required by a reversible machine producing the same amount of cold gas at the same temperature divided by the work required by Ranque's tube using a reversible isothermal compressor. This gives

$$\eta = \frac{A_c}{A_1} = \frac{W_c a_c}{W_1 a_1} = \sigma\frac{a_c}{a_1} = \sigma\,\frac{\dfrac{T_c}{T_o} - 1 - \ln\dfrac{T_c}{T_o}}{\dfrac{k-1}{k}\ln\dfrac{p_1}{p_c}} \qquad (9)$$

In Figure 5 are shown Hilsch's data for his small tube and in Figure 6 is shown the efficiency of this tube computed from those data using Equation (9). Since the air was actually somewhat humid, the availability was slightly higher than that credited to it. It is seen that the efficiency is in the neighborhood of 1 percent. For larger tubes under ideal conditions the efficiency may reach 2 percent.

If the hot gas is credited as to temperature availability, the efficiency of the tube of Figure 5 is of the order of 2 percent; if the hot gas is credited as to temperature and pressure availability, the efficiency ranges between 10 percent and 20 percent.

Conclusion

To air condition a passenger automobile requiring 7000 Btu per hr of refrigeration, assuming a coefficient of performance of 0.13, would require an ideal horsepower of 21. This would call for a larger tube which, also operated at a somewhat higher pressure, might bring the actual horsepower to a minimum of 20. For a domestic refrigerator requiring 400 Btu per hr, using the same assumptions, the ideal horsepower is 1.2.

V. Shaefer and I. Langmuir, in their celebrated work on the artificial causing of snow, have found that the ice crystals in the cold air act as very good seeds for the snow. Where a gas is throttled to obtain liquid by the Joule-Thomson effect, the Ranque tube would produce more liquid per pound of gas supplied to it. Some of the liquid would collect on the tube wall near the nozzles and should be drained off through proper openings before it is re-evaporated. However, in most cases where such throttling is done on an appreciable scale, the products pass through heat exchangers which maintain the process, and it would produce no advantage thermodynamically to use the tube.* Hilsch uses the tube in a small gas liquefier in place of an expansion engine or other preliminary cooling means; advantages of rapidity of starting are cited.[7]

While Ranque's tube is finding a few isolated uses, we have seen that it cannot serve in installations of size where power consumption is a consideration. For refrigeration in minute quantities or for very occasional use where stored compressed air is available, it is suitable. It remains one of the most remarkable inventions of the century.

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* Private communication from Prof. S. C. Collins.

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APPENDIX

Theories of Ranque and Hilsch

In 1933[1] Ranque wrote, "The compressed external layers only have a low velocity while the expanded central layers have the greatest part of their energy in kinetic form and rotate at a very high angular velocity. In incompressible fluids, to a first approximation, the angular velocity of each layer is inversely proportional to the square of its diameter.*

"It follows that such a distribution of velocities gives rise to considerable friction between one layer and the next, such that if the layers are long enough, an equilibrium will tend to be established in which *all the layers acquire the same angular velocity.* Therefore there is a *centrifugal migration of energy,* the central layers giving their velocity to the external layers." (Italics Ranque's.)

In 1946[4] Hilsch wrote, "The air passing through the orifice has been expanded in the centrifugal field from the region of high pressure at the wall of the tube to a low pressure near the axis. During this expansion it gives a considerable part of its kinetic energy to the peripheral layers through internal friction. The peripheral layers then flow away with increased temperature. . . . If there were no internal friction, the velocity of the air would increase to a supersonic value in the expansion from the circumference to the axis, sufficient pressure ratio being available. The internal friction is particularly effective in this range of velocities. It causes a flow of energy from the axis to the circumference by trying to establish a uniform angular velocity across the entire cross-section of the tube."

Flow in the Vortex

The flow possesses rotational symmetry so that conditions are independent of the angular coordinate θ. The basic equations are the momentum or Navier-Stokes equation, the energy equation, the continuity equation, and the equation of state. These, combined with the boundary conditions, could give the solution for everything that happens in the tube if the prodigious mathematical task could be carried out. Since this is too difficult with friction and heat transfer, we investigate the situation that would exist if the effects of friction and heat transfer were negligible.

Assuming steady flow, perfect gas, no friction, and no heat transfer, the flow will be irrotational, reversible, and adiabatic, and the vortex will be a free vortex ($\omega \sim 1/r^2$). We may then write equations of energy, state, isentropic change of state, constant angular momentum, continuity, definition of velocity potential ϕ, and definition of stream function ψ. The last three of these equations are as follows:

$$\frac{\partial\ (\rho u r)}{\partial r} + \frac{\partial\ (\rho w r)}{\partial z} = 0 \tag{10}$$

$$u = \phi_r, \qquad v = \frac{\phi_\theta}{r}, \qquad w = \phi_z \tag{11}$$

$$u = \frac{\psi_z}{\rho r}, \qquad w = -\frac{\psi_r}{\rho r} \tag{12}$$

where the subscripts on ϕ and ψ indicate partial differentiation . Equation (12) is a consequence of Equation (10); the failure of the tangential velocity v to appear means that ψ = constant will not be a streamline, but a stream-surface of revolution.

If all of the above-mentioned equations are solved for ϕ and ψ with the Mach numbers M as parameters, the result is the following partial differential equations of the second order:**

$$\phi_{zz}(1-M_z^2) + \phi_{rr}(1-M_r^2) - 2M_zM_r\phi_{zr} + \frac{\phi_r}{r}(1+M_\theta^2) = 0 \tag{13}$$

$$\psi_{zz}(1-M_z^2) + \psi_{rr}(1-M_r^2) + 2M_zM_r\psi_{zr} - \frac{\psi_r}{r}(1+M_\theta^2) = 0 \tag{14}$$

where $M_z = w/c$, $M_r = u/c$, $M_\theta = v/c$, c^2 = (sound velocity)$^2 = kgRT$

It is seen that if either M_z or M_r becomes greater than unity, certain terms change sign. This means that the flow would go over to a supersonic type and shock waves or limit lines could then appear. However, in Ranque's tube it seems certain that, because of the small flow density in the radial and axial directions at all points, M_z and M_r are well below unity everywhere except that M_z may approach unity in the cold-gas orifice. The latter event is unimportant in the present consideration.*

If the tangential Mach number M_θ becomes greater than unity, there are no changes in sign, and therefore the character of the flow remains unchanged. The occurrence of supersonic tangential velocity would not be accompanied by the limitations of ordinary supersonic flow, and a continuous transition from subsonic to supersonic flow or the reverse would be possible without hindrance. Under typical operating conditions the value of M_θ at the nozzle exit is near unity and M_θ would be expected to reach high values such as 2 or more at smaller radii; it would be indicated by Equations (13) and (14) that this is free to occur. Hilsch speaks of the particular effectiveness of internal friction at high velocities; this thought may apply to the supersonic vortex in that small shock waves or shocklets may come and go, without very much changing the velocity distribution but increasing the turbulence and shearing friction. No steady shock is possible if symmetry is preserved.

The irrotational or frictionless type of flow cannot, of course, penetrate very near to the axis, because of the energy limitation. As in all vortices, on and near the axis the vortex will be approximately forced (ω = constant).

If all the Mach numbers in Equations (13) and (14) are set equal to zero, we have incompressible flow. The vortex disappears entirely from the equations. Therefore the vortex only influences the shape of the stream-surfaces of revolution to whatever extent they are shifted at the boundaries. The last terms, ϕ_r/r and ψ_r/r, account for the geometrical squeezing of the flow at the center of the tube due to the circular shape, since gas proceeding toward the center encounters a decreasing cross-section. When the compressible vortex is superposed, with M_θ taking on finite values, the magnitude of the squeezing terms is simply increased. This is due to the density gradient caused by the vortex—i.e., to compressibility.

The stream-surfaces are, then, squeezed outward by both the geometrical effect and the density effect, resulting in a rapid increase in the spacing of the stream-surface toward the center of the tube. From the continuity equation the distance between stream-surfaces very close together varies as:

$$c_\psi \sim \frac{1}{r\rho\sqrt{u^2 + w^2}} \tag{15}$$

Friction in the Vortex

In general at any point in a fluid there may exist six shear stresses on an elementary volume of the fluid: two mutually perpendicular shear stresses on each of three mutually perpendicular faces. Using cylindrical coordinates for Ranque's tube, two of these shears vanish because of rotational symmetry. These are the two shears on a surface perpendicular to the tangential velocity. The remaining four shear stresses which can exist in the tube are shown in Table 1, evaluated for laminar flow. For turbulent flow the viscosity μ simply becomes a much larger number. In Table 1 the first subscript indicates the perpendicular to the plane of the shear stress, while the second indicates the direction of the shear stress.

Table 1. Work or Power Flux

Work flux component	*Associated shear stress (laminar value)*	*Magnitude (laminar value)*	*Mechanism*	*Direction of work flux*
$P_{r\theta}$	$\tau_{r\theta} = \mu r \frac{\partial\omega}{\partial r}$	$dP_{r\theta} = -\mu r^2\omega \frac{\partial\omega}{\partial r} dS_r$	one cylinder turning within another	radial
$P_{z\theta}$	$\tau_{z\theta} = \mu r \frac{\partial\omega}{\partial z}$	$dP_{z\theta} = -\mu r^2 \frac{\partial\omega}{\partial z} dS_z$	one transverse plane turning against another	axial
P_{rz}	$\tau_{rz} = \mu \frac{\partial w}{\partial r}$	$dP_{rz} = -\mu w \frac{\partial w}{\partial r} dS_r$	one cylinder slipping axially within another	radial
P_{zr}	$\tau_{zr} = \mu \frac{\partial u}{\partial z}$	$dP_{zr} = -\mu u \frac{\partial u}{\partial z} dS_z$	part of a transverse plane slipping radially against another plane	axial

In cylindrical coordinates, the shear stress $\tau_{r\theta}$, caused by the turning of one cylinder inside another, is not given by $\tau_{r\theta} = \mu \frac{\partial v}{\partial r}$. That this would be incorrect is seen in the case of rotation as a solid body, where $\tau_{r\theta}$ must be equal to zero while $\partial v/\partial r$ is not equal to zero. The correct relation is

$$\tau_{r\theta} = \mu \left(\frac{\partial v}{\partial r} - \frac{v}{r} \right) = \mu r \frac{\partial \omega}{\partial r} \tag{16}$$

In Ranque's tube this is by far the largest shear stress and it is the primary cause of the energy migration.

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* Ranque need not have limited this statement to incompressible flow.

** Sauer[8] gives for spatial flow in cylindrical coordinates a general equation for ϕ which reduces to Eq. (13) for rotational symmetry. For axially symmetric spatial flow without vortex he gives equations for ϕ and ψ which amount to Eqs. (13) and (14) with M_θ missing.

* In a plane vortex ($\partial/\partial z = 0$), because of the restraint of the sides, a subsonic radial Mach number is forced to increase with decreasing radius in order to pass the flow radially, giving a limit line where M_r becomes unity. This happens whether a vortex is superposed or not. In Ranque's tube the axial freedom permits the flow to spread lengthwise.

This shear stress is constantly at work to convert the free vortex into a forced vortex, but it is able to do so only in a surprisingly indirect manner. Suppose that a free vortex exists in some region in the tube. Then in this region, $\omega = \text{constant}/r^2 = C/r^2$, and $\delta\omega/\delta r = -2C/r^3$. Then from Equation (16), $\tau_{r\theta} = -2\mu C/r^2$. Thus the torque acting on a cylindrical surface of radius r and unit axial length is

$$Q = -2\mu\frac{C}{r^2} \times 2\pi r \times 1 \times r = -4\pi\mu C \tag{17}$$

This is a constant. Therefore, on any thin or thick cylinder of the gas, there is a torque acting on the outer surface, and an equal and opposite torque acting on the inner surface. The cylinder experiences no net torque and consequently the friction cannot either slow it down or speed it up. The solution to this state of affairs is explained by Kassner & Knoernschild. The innermost and outermost layers of the vortex—those at the axis and tube wall, respectively—do not have this balanced torque because they are at the boundaries. Therefore the friction can act to slow down the innermost layers and speed up the outermost layers while the interior layers undergo no change in speed. Given sufficient time and space, a forced vortex will progressively eat its way into the free vortex from the boundaries until the two advancing fronts meet and the free vortex has been completely destroyed.

K. & K. have analyzed the turbulence in the tube and found that the turbulent shear stress is everywhere a constant times the laminar shear stress. In other words the viscosity is simply increased to a certain high but constant value. Therefore the same deductions made from Equation (17) for laminar flow should hold, at least approximately, for turbulent flow. K. & K., by estimating the shear stress, draw a picture showing how the regions of free and forced vortex might appear in Ranque's tube. An idea of their picture can be obtained as follows: In Figure 3, locate a point halfway between the letter *b* and the wall of the tube. From this point draw two straight lines, one to the lefthand corner of the nozzle exit and the other to the corner of the orifice near the letter *a*. The triangle so drawn is a cross-section of the pointed annulus in which the vortex is free; everywhere else it is mainly forced.

Pending experimental measurements, the shape and length of the free vortex envisioned according to this theory must be considered quite indefinite.

Flow of Work and Heat in Ranque's Tube

In a fluid where there is a gradient of velocity causing shearing friction, if we imagine a surface lying parallel to the flow and perpendicular to the velocity gradient, the fluid on one side of the surface is doing work on the fluid on the other side. If the shear stress at the surface is τ and the velocity of the fluid is V, the rate of flow of work (power) per unit area of surface is

$$P = \tau V \text{ ft-lb per sec ft}^2 \tag{18}$$

The work flows perpendicularly across the surface from the more rapidly moving fluid to the less rapidly moving fluid. The amount of heat generated in the surface itself is infinitesimal, because the surface is infinitesimally thick and the difference in velocity across it is infinitesimal. Therefore although some of the work may be dissipated into heat somewhere beyond the surface, all of the work goes across the surface.

With each shear stress in Ranque's tube there is associated a rate of flow of work according to Equation (18). These are shown in Table 1. In Table 2 are shown the two possible heat flows caused by gradients of static temperature.

Table 2. Heat Flux

Heat flux component	*Magnitude (laminar value)*	*Direction of heat flux*
q_r	$dq_r = \kappa\frac{\delta T}{\delta r}dS_r$	radial
q_z	$dq_z = \kappa\frac{\delta T}{\delta z}dS_z$	axial
	$dS_r = 2\pi r\,\lvert dz\rvert$	$dS_z = 2\pi r\,\lvert dr\rvert$

In Tables 1 and 2 the signs are taken as if the origin of coordinates were a thermodynamic system: work flowing away is positive, heat flowing away is negative. The work always flows from the more rapidly moving gas to the less rapidly moving gas.

Since the two predominant energy fluxes are the centrifugal work flux P_{r_θ} and the centripetal heat flux q_r, let us find their relative magnitudes in a free vortex under the simplifying assumption that the gradients of the radial and axial velocities u and w are small so that T will be a function of the tangential velocity v only. Then the energy equation is

$$v^2 + 2gJc_pT = \omega^2r^2 + 2gJc_pT = \frac{C^2}{r^2} + 2gJc_pT = \text{constant} \tag{19}$$

The rate of centrifugal work flux across an elementary cylindrical area dS_r is, from Equations (16) and (18),

$$dP = \tau v dS_r = -\mu r^2\omega\frac{\delta\omega}{\delta r}dS_r = 2\mu\frac{C^2}{r^3}dS_r \tag{20}$$

The rate of centripetal heat flux across the same area is, from Table 2,

$$dq = \kappa\frac{\delta T}{\delta r}dS_r = \kappa\frac{C^2}{Jgc_pr^3}dS_r \tag{21}$$

Therefore the ratio of the work flux to the heat flux is constant throughout the vortex, and over any area the total work flux is related to the total heat flux by the equation

$$\frac{\text{Work Flux}}{\text{Heat Flux}} = \frac{P}{Jq} = 2\frac{\mu g c_p}{\kappa} = 2N_{\text{Prandtl}} \tag{22}$$

This equation applies either to rates or to total quantities of energy flow. In the corresponding classical problem in Cartesian coordinates, the coefficient of the Prandtl number is unity instead of 2. Equation (22), then, reveals a peculiar property of the free vortex.

To adapt Equation (22) to turbulent flow, we only need to affix asterisks to μ and κ so that they will now represent the turbulent viscosity and the turbulent thermal conductivity:

$$\frac{\text{Work Flux}}{\text{Heat Flux}} = 2\frac{\mu^* g c_p}{\kappa^*} = 2N^*_{\text{Prandtl}} \tag{23}$$

Equation (23) is a basic equation for Ranque's tube.

Is the turbulent Prandtl number equal to the laminar Prandtl number? By the kinetic theory of gases and the theory of turbulence, they should be equal because turbulence is considered to be the same process as molecular diffusion, but on a much larger scale. Furthermore, a number of experiments on various gases under various conditions of Cartesian and polar flow, recently correlated, corroborate this theory; the turbulent Prandtl numbers lie in the range 0.7 to 0.85.*

For the laminar Prandtl number McAdams ** gives the following values at 1 atm. and 212 F: for air, CO, H_2, N_2, and O_2, 0.74; for CO_2 and SO_2, 0.80; for ethylene, 0.83. These are fairly independent of temperature and pressure.

This leads to the following conclusions. The higher the Prandtl number, the better would we expect Ranque's tube to perform, other things being equal. For a Prandtl number of 1/2 we should expect no temperature change. For a Prandtl number of less than 1/2 we should expect the tube to deliver hot gas from the orifice and cold gas from the tube.

However, the range of Prandtl numbers in nature is too small. McAdams cites the semi-empirical formula,

$$N_{\text{Prandtl}} = \frac{4}{9 - \frac{5}{k}} \tag{24}$$

where $k = c_p/c_v$. Inserting the basic values of k into Equation (24), we obtain:

Gas	k	N_{Prandtl}
Monatomic	1.67	0.666
Diatomic	1.40	0.737
Triatomic	1.28	0.786

Therefore it seems unlikely that Ranque's tube could ever be caused to operate in the reverse manner. There is possibly just enough difference between the Prandtl numbers of some gases to show the dependence predicted by Equation (23).

If we assume the inner gas gives away all the possible work $c_p\,\Delta T_{\text{isentropic}}$, where $\Delta T_{\text{isentropic}}$ is defined in Equation (1), we obtain the maximum possible temperature drop of the cold gas from Equation (23):

$$\frac{\Delta T_{\text{maximum}}}{\Delta T_{\text{isentropic}}} = 1 - \frac{1}{2\,N^*_{\text{Prandtl}}} \tag{25}$$

For $N^*_{\text{Prandtl}} = 0.75$, an approximate value for air, Equation (25) gives a temperature drop ratio of 1/3. Since Ranque's tube has exceeded this performance almost by a factor of 2, we may conclude that some of our simplifying assumptions have introduced appreciable error, or that the turbulent Prandtl number in Ranque's tube is, for some reason, higher than in other types of flow.

The Problem of Predicting the Performance

To calculate the temperature drop of the cold gas by integrating the work and heat flux across the dividing stream-surface would require knowledge of the following quantities on that surface: the size and shape, the angular velocity and its gradient, the gradient of static temperature, the turbulent viscosity, and the turbulent thermal conductivity. It would, therefore, be a formidable, even hopeless, task to try to predict the experimental results accurately by this method, particularly since all these quantities would have to be determined for all conditions of operation.

If we assume that the effective portion of the dividing stream-surface is a cylinder of length l and radius r, and that a simple free vortex exists in this region, we then obtain the following formula by applying Equations (20) and (23):

$$\Delta T_{\text{cold gas}} = \frac{4\pi}{J\sigma W c_p}\, l\,\mu\, v_j^2\left(\frac{r_j}{r}\right)^2\left(1 - \frac{1}{2\,N^*_{Pr}}\right)\frac{\mu^*}{\mu} \tag{26}$$

A sample calculation may be made for Hilsch's small tube (Fig. 5) with the following assumptions: r_j = tube radius = 0.1 in., v_j = nozzle jet velocity = 1000 ft per sec, $r_j/r = 2$, $l = 6r_j = 0.6$ in., $\sigma = 1/2$, W = total flow = 18 lb per hr, $c_p = 0.24$ Btu per lb °F, $\mu = 3.2 \times 10^{-7}$ lb-sec per ft², $N^*_{Pr} = 0.75$. Assuming a Reynolds number can be assigned to the flow equal to the nozzle jet velocity multiplied by the tube diameter and divided by the kinematic viscosity, then using a kinematic viscosity of 1.3×10^{-4} ft² per sec, this Reynolds number is 128,000. Assuming the viscosity ratio μ^*/μ is the same as it would be in ordinary pipe flow in a perfectly smooth pipe at this Reynolds number, the viscosity ratio is 350. Then substituting all these quantities into Equation (26), the result is $\Delta T_{\text{cold gas}} = 20$ F. Since this is only about one-fourth of the experimental value, it would appear likely that the turbulence is a good deal more intense than was assumed.

To approach more closely to the prediction of the temperature drops, K. & K. have invented the following procedure, best understood by considering the uniflow type alone. At a certain section near the discharge end of the tube, the vortex is considered to have become completely forced (ω = constant). The axial velocity is assumed uniform and the radial velocity zero. The distribution of states across the section is assumed to be that of reversible and adiabatic changes of state of the small pockets of gas as they move in and out radially in the turbulent flow, leading to the equation p/ρ^k = constant. This, combined with the equation for centrifugal pressure gradient and the equation of state, gives an integral solution for all properties across the section in terms of three unknown parameters. To calculate these, one applies the three conditions that the total energy, the total flow, and the total angular momentum crossing the section are respectively equal to those quantities at the nozzle exit. A correction can be made for loss of angular momentum on the tube wall if desired. Having finally solved the forced vortex entirely, one separates the flow at any desired radius.

K. & K. cite sample calculations by their method which are in reasonable agreement with experimental results. The mathematical task is formidable, and it is apparently impossible to express the results in a general way and still retain accuracy.

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* Private communication from Prof. A. H. Shapiro.

** *Heat Transmission*, McGraw-Hill Book Co., New York, 1942, p 415.

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A Simply Constructed Vortex Tube for Producing Hot and Cold Air Streams

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[*Paper received 2 January 1950*]

ABSTRACT. Hilsch has demonstrated that a jet of air to which a spiral motion has been imparted can be separated into two streams, one heated by compression and the other cooled by expansion. A simplified form of Hilsch's device is described, which will produce a stream of hot air and a stream of cold air (maximum temperature 88° C., minimum temperature −20° C.) from a source of compressed air at room temperature.

Hilsch* has described the construction and performance of a vortex tube in which compressed air is admitted through a tangential nozzle into a suitably shaped cylindrical container, from one end of which the air escapes at an elevated temperature and from the other end at a reduced temperature. The performance of this remarkable device is astonishing. For example, using the simplified version described below, air admitted at 4 atm. and 15° C. emerged from the hot end of the tube at a maximum temperature of 88° C., and from the cold end at a minimum of −20° C. The effect depends on the fact that the rotating air stream produces a region of increased pressure near the wall inside the cylinder, and a region of decreased pressure near the axis. The device separates air from these two regions into two streams, one heated by compression, and the other cooled by expansion.

In his paper, Hilsch states: 'The use of a thin-walled German-silver tube and the arrangement of screws and flanges for the assembly as described earlier reduces the heat transfer between the warm and the cold part to a small value.' It occurred to the writer that heat transfer could be still further reduced by constructing the device in a material of low thermal conductivity, such as Perspex, with glass tubes to carry off the hot and cold air. Additional advantages in the use of Perspex are the ease with which it can be machined and cemented and its transparency, which allows any blockage of the device by snow or dirt to be observed without dismantling.

In the device as described by Hilsch, the air injected through the jet travels once round a spiral channel (i.e. similar to one turn of a watch-spring) before continuing to spin in a path of circular cross-section. This spiral channel is somewhat troublesome to construct and a circular channel was experimentally substituted. The change caused only a small decrease in efficiency and has made possible a very simple method of construction. Fig. 1 *a* shows the component parts of the simplified vortex tube. The five components *A* to *E* are all made from 1 in. diameter Perspex rod, faced and bored to the appropriate internal diameter in the lathe. Care should be taken to ensure that the faces are parallel. *A* and *E* are bored to the outside diameter of the glass tubing which they will hold, in this case 12·5 mm., and are both 20 mm. thick. *B* is 2 mm. thick and is bored to the same internal diameter as that of the glass tube, 9·6 mm. *C* is 2·5 mm. thick and is bored to 14 mm. internal diameter. A tangential cut 2·3 mm. wide is made in this part as shown. This will form the jet when it is cemented between *B* and *D*. *D* is 2 mm. thick and is bored to 4·3 mm. internal diameter. The bore of each part is carefully polished with Perspex No. 1 polish, as is the jet slot. The parts are then cemented together with chloroform in the order shown (i.e. *A* to *B*, *B* to *C*, *C* to *D*, *D* to *E*). When the resulting cylinder is firmly cemented, a flat surface is filed or milled on it parallel to its axis and at right angles to the jet. The flat surface is cut

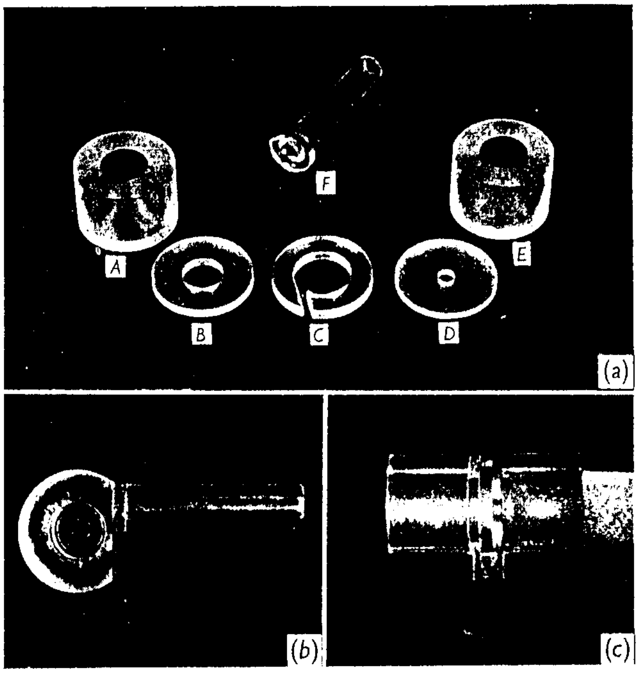

Fig. 1. Vortex tube and components: (*a*) component parts; (*b*) end view showing inlet tube; (*c*) side view of tube in operation, showing disposition of component parts

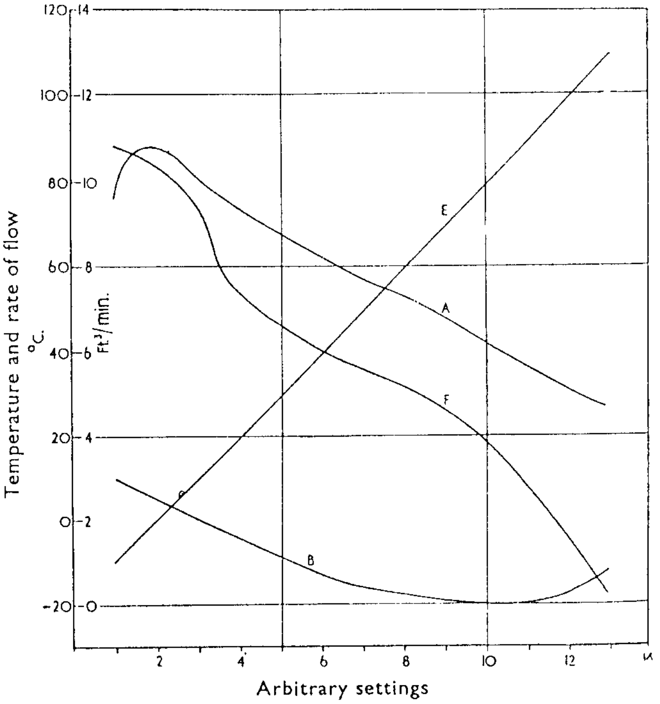

Fig. 2. Experimental results showing performance of vortex tube: *A*, °C. (hot); *B*, °C. (cold); *E*, ft.³/min. (hot); *F*, ft.³/min. (cold)

sufficiently deep to form an emplacement on to which the inlet tube *F* may be cemented (Fig. 1 *b*). The inlet tube has a bore of 5 mm. diameter and care must be taken to cement it coaxially with the jet. Lengths of glass tube are now fitted into each end of the Perspex assembly; they should fit reasonably tightly to avoid air leakage.

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* Hilsch, R., *Rev. Sci. Instrum.* 18, p. 108 (1947).

The 'hot' tube, fitting into *A*, should be at least 50 cm. long and must have a short piece of stout rubber slipped over its open end. This is fitted with a screw-clip and acts as a variable 'throttle'. The length of the 'cold' tube is not important, but may, if convenient, be made the same length as the hot tube. The inlet tube *F* is connected to a compressed air line and the air is turned on. If the screw-clip on the 'hot' tube is then progressively tightened, the issuing air will become hotter and hotter as its flow is diminished. When diminished too much the temperature of the air falls owing to the heat loss becoming greater than the rate at which heat can be supplied by the air. The temperature of the air in the 'cold' tube falls as the air flow in this tube is decreased (by allowing a greater flow through the 'hot' tube). Here also, if the air flow is decreased too much, more heat is absorbed from the surroundings than can be compensated by the small flow of cold air and the temperature will rise after passing through a minimum. Fig. 1 *c* shows the device in operation, with a heavy deposit of frost on the 'cold' tube.

The performance of the vortex tube described above when fed with air at 4 atm. is shown in Fig. 2. It will be seen that the temperature and volume of air from both 'hot' and 'cold' tubes are plotted against a series of throttle settings chosen to give a convenient set of air-flow readings from the 'hot' tube. The extremes of temperature reached will be seen to be 88° and −20° C.

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A new, sound way to refrigerate

The steam engine, the internal combustion engine, the Stirling engine, the refrigerator and the heat pump are all examples of heat engines. At a recent meeting of the Acoustical Society of America, physicist John C. Wheatley of the Los Alamos National Laboratory reported a new addition to the list: the "acoustic heat engine," which uses sound waves to generate temperature changes. This acoustic device is an application of general principles that open up the possibility of a new class of heat engines, says Wheatley.

Wheatley's acoustic engine consists of a 1-meter-long tube, about 3 centimeters in diameter and closed at one end. A stack of fiberglass plates, about 10 centimeters long and spaced a millimeter apart, sits inside the tube (see diagram). The tube is filled with a gas (helium, in this case) and closed by a piston that can vibrate at acoustic frequencies, from 150 to 1,000 cycles per second. If the plates were not present, the piston's vibrations would simply cause the tube walls to heat up. When the plates are inserted and the piston vibrates at such a frequency that acoustic resonance occurs, "then something really remarkable happens," says Wheatley. One end of the plates rapidly becomes cold while the other heats up. In a few minutes, the temperature difference may be 100°C or more.

Wheatley says that as the gas in the tube oscillates across a plate's surface, heat flows out of one end of the plate, through the gas along the plate's surface and back into the other end. "As the temperature difference develops between the two ends, the amount of heat that flows decreases, and there is a more-or-less limiting value of the temperature difference, depending mainly upon the gas," he says.

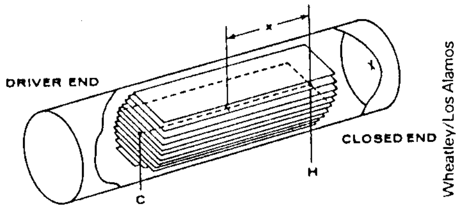
"The plate material doesn't matter," says Wheatley. "However, if you want to develop a longitudinal gradient, then it's important that you not short out the effect by having all the heat you're pumping in the gas come skidding back down this solid material." A thermal insulator like fiberglass works better than a metal.

Wheatley acknowledges that an enormous amount of prior work has been done on thermal acoustics. For example, "creating sound from heat is a most exceedingly ancient topic," he says. What's new is the insertion of plates inside a tube and the potential use of such a device as a kind of refrigerator (by utilizing the cold end).

More significant, Wheatley says, is that the principles embodied in the acoustic heat engine may apply to a broad range of physical systems. Instead of a gas and plates, any two thermally active materials in reciprocating motion with respect to each other can be used. The effect would occur as long as several thermodynamic conditions are met. These general principles can apply, for example, to magnetic solids oscillating in a magnetic field. "They may apply to other physical processes that on the surface aren't remotely connected with the general thought of heat engines," says Wheatley.

What's next? "Thought," says Wheatley. "We're going to do a quantitative study of heat flow, work flow and so on in this engine. Then we are going to try to think of ways of making it into a cooling engine."

"It's really important to note that we do not have any kind of a practical device," Wheatley cautions. It may be possible to achieve cooling to cryogenic temperatures, but several more ingredients have to be added. Meanwhile, Wheatley and his group have submitted a paper on the basic ideas behind the acoustic heat engine to PHYSICAL REVIEW LETTERS. —*I. Peterson*



Wheatley/Los Alamos

Refrigerating Engineering October 1950

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Fluid Action in the Vortex Tube

A Report of Experiments in Which a Colored Liquid Is Introduced into a Vortex Tube To Enable Visual Study of the Strange Separation of Hot and Cold Air

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The perplexing phenomenon of the vortex tube, also called Ranque's tube or Hilsch's tube, which produces hot and cold air when air under pressure enters the tube tangentially, continues to intrigue various investigators. In this article the author describes the visual effects that were seen in uniflow and counterflow vortex tubes when a trace of colored water was injected into the incoming air stream. For a thorough discussion of the operation of the vortex tube, the reader is referred to the article "Ranque's Tube" which appeared in the May 1950 issue of REFRIGERATING ENGINEERING.

IN 1931 Georges Joseph Ranque developed a method of separating a stream of compressed gas into two streams of gas, at a lower pressure, having a definite temperature differential. The work of Ranque lay undeveloped until Rudolph Hilsch became interested in the vortex tube in relation to his low temperature experiments. The investigations of Hilsch were brought to the United States by Robert Milton of Johns Hopkins University. Since then much independent investigation has taken place in colleges and industry.

The vortex tube is a device that separates an incoming stream of compressed gas into separate streams of hot and cold gas, both at a lower pressure. This result is obtained by causing a vortex in a tube of circular cross section. The vortex is created when the gas enters into the tube on a tangent to the cross section at a high velocity. A study of the cross section of the vortex shows the center portion of the vortex to be at a lower temperature and pressure than the outer portion. The separation of the center and outer portions is necessary if the hot and cold streams are to be obtained. This is done by two methods, the uniflow and the counterflow.

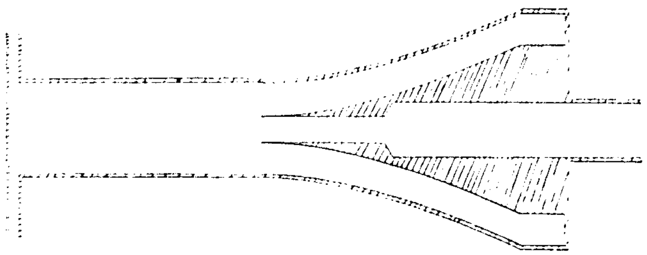

Fig. 1. The uniflow type of vortex tube.

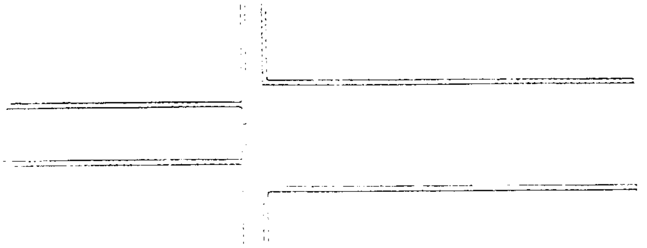

Fig. 2. The counterflow type of vortex tube.

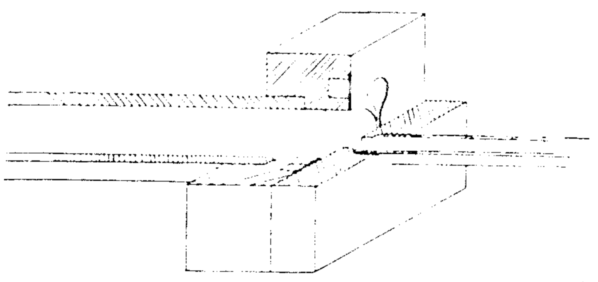

Fig. 3. A working model of the counterflow vortex tube.

The uniflow method of separation is shown in Figure 1. The center of the vortex is separated by using a sampling tube in the middle to remove the cold gas, and a concentric outer tube to take off hot gas.

The counterflow method of separation is shown in Figure 2. The center of the vortex is forced out the small tube on the left while the outer portion is allowed to flow out the larger tube on the right.

The hot stream tube must be kept at a higher pressure than the cold stream tube, since the center portion of the vortex is at lower pressure. If the cold tube exhausts to atmosphere, it is necessary to throttle the gas flow in the hot tube.

By holding the physical dimensions of the tube and inlet gas pressure and temperature constant, the amount of flow through the cold tube may be varied by adjustment of the hot tube throttling. Since the vortex tube obeys the laws of thermodynamics, this throttling also varies the gas temperatures with the gas flows. Two interesting cold gas points may be found, the lowest temperature and the greatest refrigeration.

The flow pattern of the gas in the vortex tube has come to be a point of discussion in the analysis of the tube. The actual facts are unknown to date; however, research is gradually reducing unknown factors.

It is easy enough to imagine what happens at the nozzles when the gas is inducted into the tube. However, this is not quite as easy as it looks when considerable thought is given to the action within the tube.

C. D. Fulton Jr. of the Massachusetts Institute of Technology first suggested the use of multiple nozzles rather than the single nozzle and spiral chamber used by Hilsch. This thought was probably motivated by the observations of other students at M.I.T. While Mr. Fulton was at the General Electric Company he made several tests with eight and sixteen nozzles.

The statements of Mr. Fulton (Ranque's Tube, REFRIGERATING ENGINEERING, May, 1950) caused the writer to become interested in the location of the center of the vortex near the nozzles. The writer has built a vortex tube of plexiglas and taken photographs of a trace of colored water injected into the inlet air Figure 3 shows this vortex tube.

Figures 4, 5, 6, and 7 show the vortex tube operating with one, two, four, and eight nozzles. The small round dark spots on the outer periphery of the tube are inlet holes to the concentric manifold connecting the inlet pipe to the nozzles.

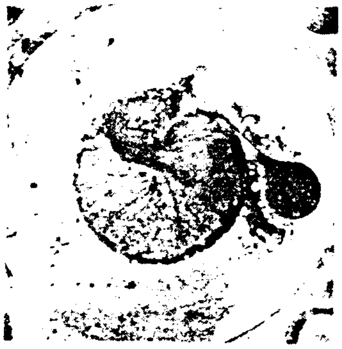

Fig. 4. End view of a single-nozzle uniflow vortex tube.

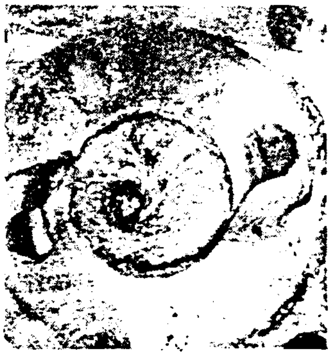

Fig. 5. End view of a two-nozzle uniflow vortex tube.

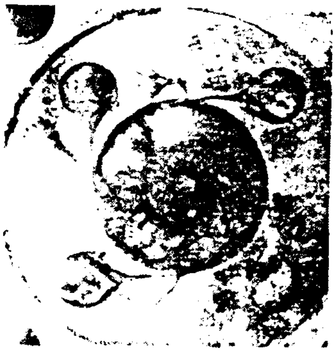

Fig. 6. End view of a four-nozzle uniflow vortex tube.

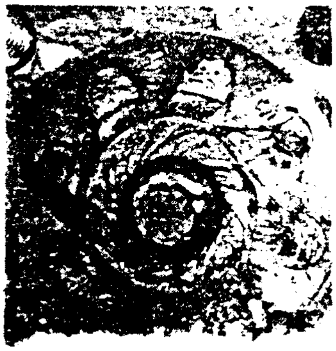

Fig. 7. End view of an eight-nozzle uniflow vortex tube.

Figure 4 shows the pattern of a vortex tube with a circular cross section and a single nozzle. The center of this pattern is seen to be almost on the outer periphery of the vortex tube. This single nozzle does not give a satisfactory pattern.

Figure 5 shows the case of a vortex tube having two nozzles. The center of this vortex pattern is still not in the middle of the tube. However, it is considerably better than the single nozzle case.

Figure 6 shows the vortex tube with four nozzles. The center of this pattern is located fairly well in the middle of the tube.

Figure 7 illustrates the case of eight nozzles. This pattern is well centered; however, it does not appear much better than the four-nozzle case. This would indicate that the four-nozzle vortex tube sufficiently centers the gas pattern near the nozzle. It should be noted that the photographs show only the pattern of the gas in the near vicinity of the nozzle.

The action of the gas as it passes down the tube is in a helical pattern. As the gas traverses the counterflow tube toward the hot tube valve, the pitch increases. Figure 8 shows the colored water trace of the gas as it travels down the tube. Actually the eight-nozzle test plate was used but, due to the inlet air arrangement, not all of the nozzles were supplied with fluid. This photograph was taken when the hot tube valve was open and air was entering the cold tube. The water was thrown to the inside surface of the tube and formed in the patterns of the pressure distribution of the gas.

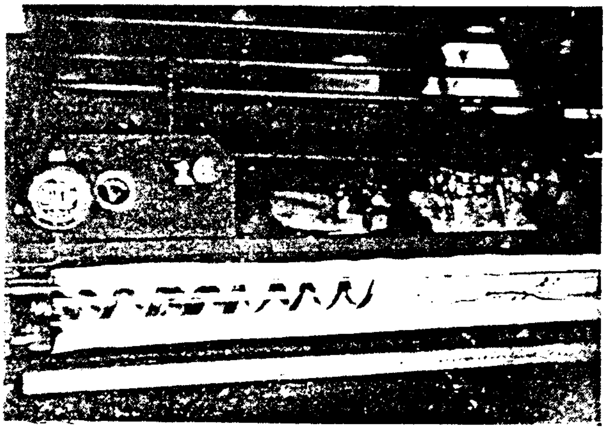

Fig. 8. Colored water trace of gas in a counterflow vortex tube.

When the hot tube valve is slowly closed down, the gas starts to flow through the cold tube. As the gas starts to flow out the cold tube, a strange unsuspected phenomenon takes place. A round ring of fluid completely encircling the tube starts in the vicinity of the valve and travels up the tube as the valve is closed down. This ring is accompanied by a loud whistle of increasing pitch as the valve is shut down. The ring comes to a stable position at the point of full closed on the hot tube valve.

A photograph of this phenomenon is shown in Figure 9. The colored water is shown as it forms on the bore of the tube. The pool of water on the right shows little activity as the hot gas fraction flows over it. Other observations show a helix until the colored water reaches the ring.

No real theory has been developed for this "end effect." The writer believes this is some sort of turning point for the gas as it flows toward the cold tube orifice. If this is true, it would indicate a high degree of mixing of the hot and cold portions of the gas.

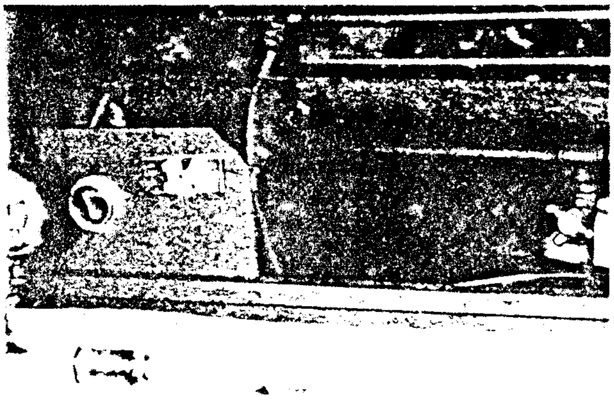

Fig. 9. "End effect" in a counterflow vortex tube.

This would indicate that the uniflow type of vortex tube would be advantageous over the counterflow type.

Comparatively little investigation has been done on the uniflow type of vortex tube, even though this type of tube seems to be more promising than the counterflow. This will probably be determined when more investigations are made on the vortex tube.

bore arm of the tube communicates only with the center of the whirlpool and therefore draws off the cold air. The longer, larger-bore arm draws off the rest of the air —the hot molecules left at the periphery of the whirlpool.

Varying ratios of hot and cold air can be obtained by adjusting the stopcock at the warm-air end. The tube is capable of simultaneous temperatures of plus 106° and minus 56° F. At "hot" adjustment it can produce up to 350° F.

While the fundamental scientists have been arguing, the applied scientists of the refrigerating industry have been doing some experimenting of their own. Industry overlooks no bets that may involve discarding methods laboriously and expensively developed. At least three big concerns have had a look at the tube. At Westinghouse, Gaylord W. Penney, manager of the Electrophysics Department, has built a model three times the diameter of the original tube. He reports:

"Not efficient for refrigeration. The tube's highest utility is as a laboratory device. For instance, it could be used as a cold trap for mercury. It could also be used to cool gases, such as helium, prior to liquefaction."

Physicists of other companies, together with those of the National Bureau of Standards, generally agree that the Hilsch tube has no particularly rosy industrial future for the time being. They point out that while the tube gives 15 to 20 times more cooling than the ordinary laboratory method—expansion of gas through a nozzle under the Joule-Thomson principle—it has refrigerating efficiency of only 20 percent, as against 70 percent in household refrigerators and close to 90 in larger cooling installations.

They are loath, however, to disregard the provocative commercial possibilities of the tube. A more highly refined model might attain the thermodynamicists' dream of a double-purpose unit, on which one switch produces heat: another, cold.

Maxwell and his intelligent demon are probably smiling quietly at some astral laboratory bench. END

The Ranque–Hilsch Effect

Researchers at the University of Tennessee Space Institute have emerged with a radical explanation of the Ranque–Hilsch effect, a spectacular and mysterious separation of air in swirling motion into hot and cold streams, and have traced the cause to a whistling sound present in the vortex flow.

In 1933, Georges Ranque, a French engineer, uncovered a striking phenomenon: when compressed air was introduced into a pipe through tangential injection holes (imparting a swirling motion) the temperature near the tube centerline became freezingly cold, while the temperature near the tube periphery exceeded its inlet value. This temperature separation took place without the aid of any external mechanical device. The effect became popularized later through a paper by Rudolf Hilsch, a German scientist. The device which became known as the Ranque–Hilsch tube or vortex tube is now commercially available; it is routinely used as a refrigerator without a motor or other moving parts for such purposes as machining operations and protection of electrical equipment, since even with inlet air at room temperature and at readily attainable pressure of a few atmospheres, cold air below 0°F can easily be obtained.

In spite of the simplicity of the Ranque–Hilsch tube, the mechanism of temperature separation has been a matter of long-standing dispute. Despite many attempts, the process eluded an adequate explanation; some attributed this, in a lighter vein, to Maxwell's demon, an imaginary being invented in the nineteenth century by James Clerk Maxwell. This microscopic demon would divide up the air by storing faster-moving molecules in one compartment and slower-moving molecules in another.

It had now long been recognized that a high-intensity whistling sound—the so called vortex whistle—emanates from vortex flow in general and from the Ranque–Hilsch tube in particular. Although previously regarded merely as an annoying by-product of such swirling flows and nothing more, the Tennessee scientists hypothesized that this vortex whistle was, in fact, the main cause of the Ranque–Hilsch effect.[1]

In order to verify this, they installed acoustic suppressors, specially designed and tuned to the frequency of the vortex whistle, on a Ranque–Hilsch tube. When the pitch of the vortex whistle, which increases as the flow through the tube is increased, hit the tuned frequency, the sound level suddenly tumbled by 25 dB, changing from an ear-splitting whistle to a muffled hiss. At that very instant, the tube centerline temperature, which had gone down to as low as − 32°F, immediately leapt upwards to + 33°F (a temperature jump of 65°F), while the temperature near the tube periphery plummeted by 10°F. This supported the theory that the vortex whistle was indeed the main cause of the Ranque–Hilsch effect.

The sound deforms the distribution of swirl velocity in the radial direction through the mechanism of acoustic streaming. Acoustic streaming, or sonic wind, is in general a phenomenon in which sound or alternating currents of flow modify the time-averaged, or direct, currents of flow; it is, in a sense, analogous to the more familiar generation of electric heat by alternating current. Owing to this acoustic streaming induced by the vortex whistle, the flow becomes distorted in such a pattern that the swirl velocity, with a zero velocity near the centerline, continues to increase in the radial direction, like the rotation of a car tire. When the air eventually comes to rest, the higher velocity near the periphery is converted into warmer temperature while the lower velocity near the centerline is converted into colder temperature.

M. Kurosaka, University of Tennessee

Physics Today
Jan' 84

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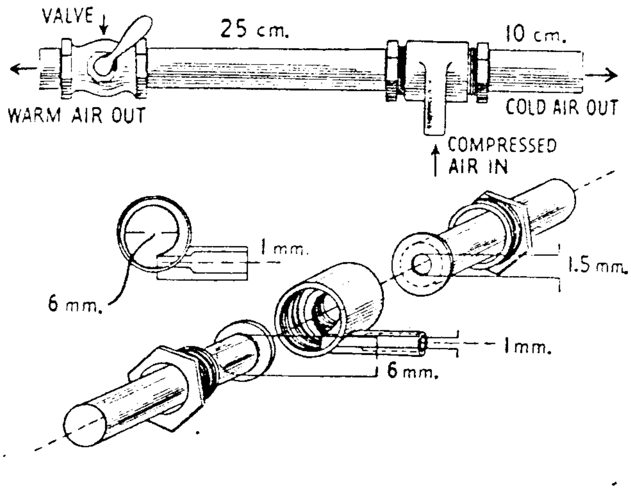
[1] M. Kurosaka, J. Fluid Mech. 124, 139 (November 1982).

INDUSTRIAL AND ENGINEERING CHEMISTRY REPORTS ON THE CHEMICAL WORLD TODAY

Technology

Maxwellian Demon at Work. The molecule sorter, discussed in the more fantastic moods of chemists and physicists, is brought tantalizingly near reality in a small T-tube now in the possession of R. M. Milton of the chemistry department, The Johns Hopkins University. By merely forcing compressed air into the center tube of this deceptively simple gadget, a stream of cold air is produced from one arm while warm air issues from the other. Milton recently visited Germany to investigate low temperature work in connection with superconductivity research. He found the apparatus in practical use in the laboratory of Rudolf Hilsch at Erlangen, where it replaced the usual ammonia precooling apparatus in a liquid air machine.

Disassembly showed the construction and functional operation of the instrument to be fascinatingly simple, as the sketch indicates.



Air forced into the center tube under pressure while the valve is fully open passes entirely out of the left-hand tube and external air is sucked into the right. However, on partially closing the valve, air at a temperature noticeably lower than that of the input passes out of the right-hand tube, while the temperature of the left effluent rises. Lowest temperatures are attained when the ratio of cold to warm effluent gas is approximately 1 to 3; maximum cooling is produced when the ratio is about 1 to 1. In a recent demonstration by Milton, using nitrogen at about 100 pounds per square inch and room temperature, frost was formed on the cold tube within a minute, while the right became warm to the touch. A similar effect was produced with hydrogen, discounting explanation on the basis of simple Joule-Thomson effect. Milton stated that with air at 10 atmospheres pressure a temperature as low as −35° C. could be produced. He stated that the temperature drop was about 50% of the isentropically ideal, and efficiency of cooling was about 20% in the present state of development.

Why does the gadget work? Hilsch's explanation, suggesting frictional loss of kinetic energy by the gas molecules in the whirling center, does not satisfy Milton; yet he and various colleagues have not deduced the answer. What is your analysis?

R.L.K.

Alcoholism in Engines. The effect of alcohol, in all of its popular dilutions, upon the human body has been the subject of considerable experimental work. It remained for the National Bureau of Standards, however, to investigate the results obtained by using alcohol and other related compounds as fuels for gasoline engines.

The evidence from their tests indicates that, although the engines enjoy a "lift" with some of the substitute fuels and give 2 to 4% more power than with gasoline, 60–70% more fuel is required. This would mean that a tank capable of holding enough gasoline to carry a vehicle 200 miles could hold only enough anhydrous alcohol (200-proof) to carry it 135 miles. With 120-proof alcohol the available mileage would be 60; with 90-proof, 40 miles; and finally with 70 proof (the content of most cordials and liqueurs), only 25 miles. In addition to these various ethanols, acetone, *n*-butanol, and ether (diethyl) blends were examined.

Four engines of different makes were equipped with electric dynamometers and special spark plugs, as well as instruments for measuring oil pressure, cooling water temperature, and fuel consumption. The spark plugs contained a hollow electrode through which an iron-constantan thermocouple was introduced to measure the electrode tip temperature.

(*Continued on page 8*)

An interpretative monthly digest for chemists, chemical engineers, and executives in the chemical producing and chemical consuming industries

Astounding Science Fiction, p. 105-110

MAXWELL'S DEMON AND MONSIEUR RANQUE

BY ARTHUR C. PARLETT

***This** is not a hoax; the article is straight fact. And it's the first time a stream of air's been induced to blow hot and blow cold at the same time!*

There was a good deal of talent wasted in the olden days when demons and jinn were imprisoned in quaint bottles, sealed with the signet of Sulayman, and cast into the sea. It has taken hundreds of years, but we at last are on our way to learning how to harness the peculiar powers of such beings. As a starter, we now take the smaller demons, imprison them in T-tubes, seal with the mark of Georges Ranque, and put them to work sorting molecules according to energy content; the hot molecules in one pile and the cold ones in another. Such, at least, was my initial—and erroneous—reaction to the operation of the T-tube whose working drawings are reproduced as Figure No. 1.

This device, shown in assembly as Figure No. 2, effects a separation of a charge of compressed air of uniform temperature into two streams. The current from the longer arm may easily reach a temperature of +200°C., while the temperature of the air stream from the tube at the right may go as low as −50°C. The conditions of the air stream to the T-tube are "normal" temperature, that is 20°C., and pressure of only a few atmospheres. It is interesting to note that the lowest temperatures are obtained when the ratio of cold air to hot, regulated by means of the valve on the hot end, is approximately 1:3.

These effects would seem to indicate that the Second Law of Thermodynamics is being successfully violated, and this in turn suggests that the demon proposed by Clerk Maxwell has been found and put to useful work!

The Second Law has often been dressed up in fancy terminology, but all it really says is (1) that water *naturally* flows down hill, not up, and (2) that heat *naturally* flows from hot bodies to cold, not the reverse. In other words, all actions that take place spontaneously tend to bring a system to equilibrium or to rest. As a secondary point, it may be noted that any process starting of and by itself—whether or not it is allowed to proceed to equilibrium—is accompanied by some loss of capacity for self-starting. This loss of capacity for spontaneous change is measured by its gain in entropy, indeed can serve as a basic definition of entropy. It will be noted that no loss of energy is involved; only the degree of availability of the energy to do useful work has been lessened.

A little reflection brings out an interesting corollary to the above: Any action that starts of itself represents a tendency to achieve a state of maximum probability, and the achievement of maximum probability is also the attainment of maximum entropy. When heat flows from a hot body to a colder one; when water flows down hill, both maximum probability and maximum entropy are being attained. Similarly with any other such action you care to name.

It seemed that Monsieur Ranque's device violated this law. Air is certainly a well mixed substance, and the number of molecules in a unit volume is fantastically large. A tremendous aggregation of gaseous molecules in completely random movement simply must arrive at the most probable configuration of complete randomness in short order. Yet the device Monsieur Ranque patented in 1934* apparently effects a separation of hot molecules from cold with consequent *loss* of entropy from the system as a whole. The action is strikingly similar to that of the demon first conceived by Clerk Maxwell.

Clerk Maxwell, remarkable physicist of the last century, recognized the statistical nature of temperature. In a gas of a certain observed temperature, a large percentage of molecules will have a kinetic energy content greatly higher than that corresponding to the thermometer reading, and a large percentage will have a much lower kinetic energy content. The temperature reading is merely the average value. That the same reading is obtained at all points in the system simply means that the system is in its state of maximum probability. The holding of a perfect bridge hand is considered a news item because the number of highly shuffled, random, chaotic arrangements are hundreds of millions of times greater than the perfectly ordered hand. Yet the probabilities in this case are based upon a deck of only fifty-two cards. Because of the unimaginable number of molecules involved, Dodge has said: "Any arrangement in which the hot molecules would predominate to permit an observable difference in temperature between any macroscopic portions of the gas is so improbable as to be nonexistent." (Dodge, B. F. *Chemical Engineering Thermodynamics.*) If only molecules could be dealt with as individuals, thought Maxwell, the laws of probability would no longer apply and the Second Law of Thermodynamics would lose its rigor.

Maxwell conceived of a tiny being, since come to be affectionately known as "Maxwell's demon", who could handle the individual molecules of a gas. He made this demon the guardian of an opening between two boxes containing a gas at uniform conditions of temperature and pressure. The demon was to control the opening by a shutter, and by judicious opening and closing of the shutter he would be able to effect a concentration of hot—high translational energy—molecules in one box and a concentration of cold—low translational energy—molecules in the other, thereby bringing about an appreciable temperature difference between the two, and hence decreasing the entropy of the system.*

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*U. S. Patent No. 1,952,281, assigned March 27, 1934. Method and Apparatus for Obtaining from a Fluid under Pressure two currents of Fluids at Different Temperatures. Inventor: George Joseph Ranque.

For some reason or other, the remarkable nature of George Joseph Ranque's invention seems to have been overlooked in this country for a good many years. It simply lapsed into the Limbo of forgotten things. But in 1946, Dr. R. M. Milton of The Johns Hopkins University visited Germany to investigate low temperature work in connection with superconductivity research. There, in the laboratory of Dr. Rudolf Hilsch, at the Physikalischen Institut, Erlangen, Germany, he found the device in the form illustrated in this article—Ranque in his patent lists a total of fifteen forms the device may take!—where it replaced the usual ammonia pre-cooling apparatus in a liquid air machine. (It is interesting to note that Dr. John R. Roebuck of The University of Wisconsin, writing in the *Journal of Applied Physics, 16,* 285 (1945) had independently presented a fully developed theory for a rotor device to serve the same purpose.) The very considerable experimentation carried out by Dr. Hilsch upon this device, and the publicity attendant to Dr. Milton's announcements has led to the general adoption of the name "Hilsch Tube" for the "Vortex Tube", as it is called by Dr. Hilsch in a recent article.*

The American Chemical Society has very kindly consented to our reproducing the working drawings of the Hilsch tube—Figure No. 1—so that any good machine shop can quite easily construct one for you. When I first saw the tube in operation, I sat with a thermodynamics textbook in my hand for some time. Then I closed it and said wearily to myself, "Well, there goes the Second Law!" For I certainly saw one stream going in and two streams coming out, and one of the outlet streams was cold and the other was hot. And if that weren't effective sorting of molecules according to energy content what was it?

It took quite a while before I realized that the major part of my difficulties arose because the following statements of the Second Law are incomplete; but these partial statements seemed perfectly reasonable to me at the time. More than one politician has found that half-truths make better propaganda than outright falsehood.

"Water won't flow up hill (two words missing)."

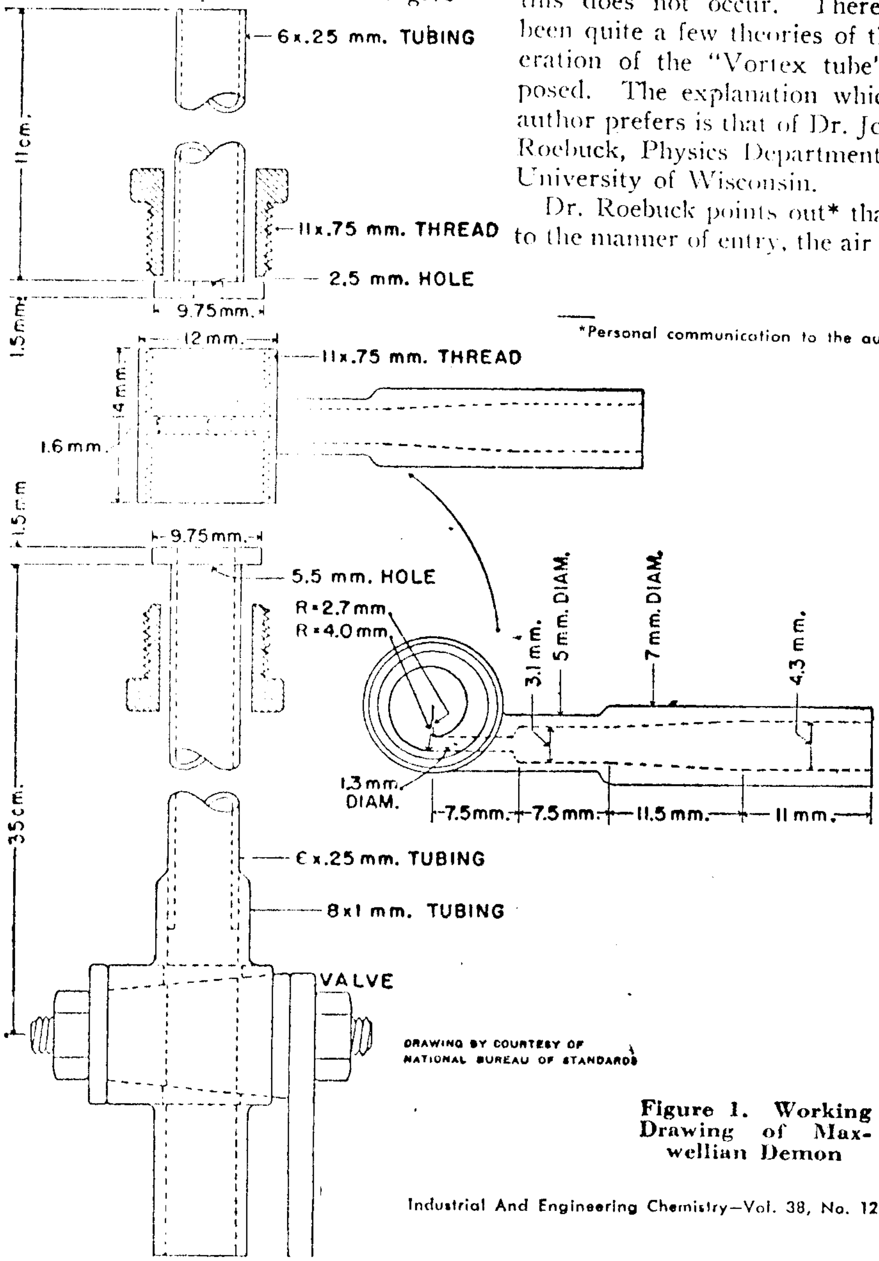
"Heat won't flow from cold bodies to hot (two words missing)."

"A system will not proceed from a more probable state to a less probable state (two words missing)."

In each case the missing two words are the same. They are "WITHOUT COMPENSATION!" In the case of the Hilsch tube, mechanical compensation for the loss of entropy by the gaseous system is provided by the compression and expansion of air. Thermodynamics doesn't give a hoot about mechanism—it is concerned only with the inherent possibilities of a given system.

But it is not even necessary to postulate that there has been a sorting of molecules, and there is much experimental evidence to show that this does not occur. There have been quite a few theories of the operation of the "Vortex tube" proposed. The explanation which the author prefers is that of Dr. John R. Roebuck, Physics Department, The University of Wisconsin.

Dr. Roebuck points out* that due to the manner of entry, the air in the



DRAWING BY COURTESY OF NATIONAL BUREAU OF STANDARDS

Figure 1. Working Drawing of Maxwellian Demon

Industrial And Engineering Chemistry—Vol. 38, No. 12

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*There is a hoary objection that even in this case the law of entropy increase would hold because the increase of entropy within the demon would more than compensate for the decrease in entropy he produced. Lewis and Randall among others, hold that before conceding this point it might be well to know something more of the demon's metabolism.

*Hilsch, R. The Use of the Expansion of Gases in a Centrifugal Field as Cooling Process. *The Review of Scientific Instruments* 18, 2, February, 1947.

*Personal communication to the author.

top of the T-tube is rotating very rapidly about the axis of the tube. This rotation sets up large centrifugal forces on the air particles outward from the axis, which produce a drop in pressure toward the axis, and a lower pressure at the axis than at the periphery. This is confirmed experimentally, as air will enter at the axial opening instead of escape unless the opening at the bottom of the whirling tube is constricted somewhat. The air coming in at the periphery forces the air in along the radii, and the falling pressure inward on the radius allows this air to expand doing work and cooling the air. It is readily shown that this work adds to the rotating energy of the body of the gas, so speeding it up. Friction with the inner surface of the tube heats up the gas next it. That is, gas drawn off at the axis has been cooled while that drawn off at the periphery near the outer wall has been warmed. Adiabatic expansion of air from ten atmospheres and room temperature to one atmosphere will cool the air by about 135° C., so there is plenty of available cooling.*

And thus the charming picture of the busy little demon sitting inside the tube with a tennis racquet in his hand, batting the molecules to right and to left as they come in, saying to himself, "There's a hot one, smack it to the right! There's a cold one, smack it to the left!" is very effectively demolished, and the Second Law continues to hold true without known exception.

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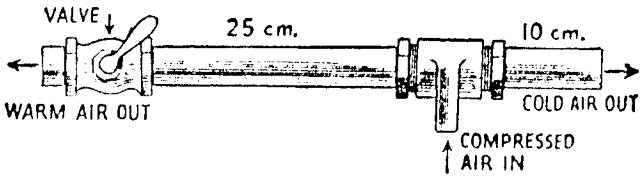
*Incidentally, Dr. Hilsch concludes his article previously referred to with the observation that ". . . . There is little probability that vortex tubes will replace the customary retrigerating machines since their efficiencies are much better in the region of small pressure ratios. There may, however, be special cases where a vortex tube would be more desirable because of simple construction (e.g., air cooling in mine shafts)."

THE END

Figure 2.



Maxwell's Demon Lives Here!

J. Fluid Mech. (1982), *vol.* 124, *pp.* 139–172
Printed in Great Britain

Acoustic streaming in swirling flow and the Ranque–Hilsch (vortex-tube) effect

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(Received 24 September 1981 and in revised form 11 March 1982)

The Ranque–Hilsch effect, observed in swirling flow within a single tube, is a spontaneous separation of total temperature, with the colder stream near the tube centreline and the hotter air near its periphery. Despite its simplicity, the mechanism of the Ranque–Hilsch effect has been [illegible] of long-standing dispute. Here we demonstrate, through analysis and experiment, that the acoustic streaming induced by orderly disturbances within the swirling flow is, to a substantial degree, a cause of the Ranque–Hilsch effect. The analysis predicts that the streaming induced by the pure tone, a spinning wave corresponding to the first tangential mode, deforms the base Rankine vortex into a forced vortex, resulting in total temperature separation in the radial direction. This is confirmed by experiments, where, in the Ranque–Hilsch tube of uniflow arrangement, we install acoustic suppressors of organ-pipe type, tuned to the discrete frequency of the first tangential mode, attenuate its amplitude, and show that this does indeed reduce the total temperature separation.

1. Introduction

The subject of acoustic streaming owes its origin to Lord Rayleigh (1884); led by Faraday's observation (1831) and the patterns in the Kundt tube, he showed that sound waves can and do generate a steady current through the action of Reynolds stresses, induced near the solid boundary by the periodic disturbances themselves. For a modern review of the subject in general we refer to Lighthill (1978*a*).

As yet, however, with the possible exception of Secomb (1978), the studies of streaming appear to have been subject to a common restriction that the fluid be initially in a state of rest.

In this paper we shall treat an acoustic-streaming problem which, by contrast, depends crucially on the motion of a gas. To be specific, *for a swirling gas within a cylindrical duct, we submit that the acoustic streaming of the gas with swirl produces the Ranque–Hilsch effect: the radial separation of total temperature.* Although it has generally – and often vaguely – been accepted that some form of unsteadiness is needed to interpret this perplexing phenomenon, the key to the explanation is precisely this unforeseen mechanism of acoustic streaming, which is induced by *orderly, periodic disturbances* within swirling flows. We shall first demonstrate this analytically, and then confirm it experimentally. But, first, a discussion on the background is in order.

Several years ago a striking phenomenon, with unusual features, accidentally revealed itself in a swirling-flow test rig called an annular cascade (Danforth 1977, private communication; Rakowski, Ellis & Bankhead 1978; Rakowski & Ellis 1978). The whole apparatus was stationary and in the shape of an annular conduit formed between inner and outer casings; therein, air, having been made to swirl by vanes, spiralled aft. A check-out test of the rig immediately disclosed the presence of a loud whistle, a pure tone, the level of which rose to a deafening intensity of 180 dB. The occurrence of this ear-splitting sound was all the more remarkable since, for one thing, none of the components of the rig was rotating; only air was in swirling motion. For another, the sound occurred even before the installation of a test cascade; except for the swirl vanes, nothing lay in the way of the flow path. Its discrete frequency increased almost proportionately to swirl, from 350 to 500 Hz; the dynamic disturbance was spinning circumferentially with the first tangential mode, accompanied by higher harmonics.

Most surprising of all, *this intense pure tone in swirling flow induced a marked distortion to the steady-state or the time-averaged components of the flow field – its 'd.c.' parts.* Accompanied by an increasingly louder whistle, the tangential velocity near the inner wall became, above a certain level, abruptly reduced to a considerable extent; the radial profile, which had remained a free-vortex type (as it was designed to be), metamorphosed into one somewhat akin to a forced vortex. At the same time the total temperature, initially uniform at the inlet and equal to 36 °C, spontaneously separated into a hotter stream of about 48 °C near the outer wall and a colder one of 28 °C near the inner wall, with a radial difference equal to 20 °C – strongly reminiscent of the Ranque–Hilsch effect.

Because of the severity of these dynamic fluctuations, which posed a serious threat to the original purpose of the rig, the inner and outer walls were provided with tuned acoustic suppressors; this did remove the unacceptable dynamic flow disturbances.

Now that the unsteady fluctuations had been eliminated, what happened to the profiles of steady flow? The answer was *that the distortion in the velocity and temperature distribution vanished. The Ranque–Hilsch effect had gone.*

Acoustic streaming appears to have somehow deformed the steady flow field, both in velocity and temperature, and if so, then this phenomenon of arresting interest affords a singular insight into the mechanism of little-understood Ranque–Hilsch effects, which we shall discuss briefly below.

We recall that in the Ranque–Hilsch tube (Ranque 1933; Hilsch 1947) or the vortex tube, compressed air enters a single straight tube through tangential injection nozzles. Once within the tube, the swirling air segregates by itself into two streams of different total temperature: the hotter air near the periphery of the tube and the colder air at the centreline, a separation effect already mentioned with regard to the annular cascade. In the conventional counterflow Ranque–Hilsch tube, the cold air is extracted from an orifice located near the inlet, and the hot air escapes from the other end. Even by closing the cold orifice, however, the air flowing in only one direction can still produce the radial separation of total temperature; this, called the uniflow type, will be adopted later for our analysis and experiment because of the simpler flow pattern.

Although the existence of total-temperature separation in Ranque–Hilsch tubes is beyond any doubt, none of the theoretical explanations so far devised appears to have found unreserved acceptance. For example, the oft-cited turbulent-migration theory (Van Deemter 1952; Deissler & Perlmutter 1960; Linderstrom-Lang 1971) fails to explain why the purported random radial migration of fluid lumps does not separate the total temperature in other swirling-flow apparatuses, which are equally turbulent and provided with the same tangential injection as the one for the Ranque–Hilsch tube; for example compare figure 3 of Batson & Sforzini (1970) with figure 4 of Hartnett & Eckert (1957). The other theories (Schepper 1951; Sibulkin 1962) appear to be open to equally serious objections.

The experimental evidence mentioned on the annular cascade compels us to turn towards acoustic streaming as an agent of the Ranque–Hilsch effect – the acoustic streaming induced through the Reynolds stresses that are caused by organized periodic disturbances rather than by stochastic, random motion.

In the literature on the Ranque–Hilsch tube, allusions to the presence of intense periodic disturbances abound. Hilsch (1947), McGee (1950), Savino & Ragsdale (1961), Ragsdale (1961) and Kendall (1962) noted the disturbances of pure-tone type, whistle or scream. In fact, Savino & Ragsdale (1961) recounted an incident where a loud screaming noise was accompanied by 6–11 °C change in total temperature, a phenomenon evocative of that experienced with the annular cascade. None of them, however, proceeded beyond the stage of making passing observations on it.

To a certain extent, the work of Sprenger (1951) foreshadows ours in its spirit. In the Ranque–Hilsch tube with its hot end closed and only its cold end open, he measured periodic disturbances by spreading lycopodium to form a Kundt pattern. However, the pattern was apparently used to measure only the wavelength of discrete disturbances, since he did not pin the Ranque–Hilsch effect down to any explicit mechanism. Rather, by appealing to the analogy of the resonance tube, he later simply suggested (Sprenger 1954) that the organized unsteadiness† might produce the energy separation. More to the point is the highly suggestive circumstance that led to the discovery of the vortex whistle by Vonnegut (1954). While engaged in experiments exploring the application of the Ranque–Hilsch cooling effect, Vonnegut (1950) had observed the presence of a pure-tone noise. Although he did not connect it with a mechanism of the Ranque–Hilsch effect, from this hint he constructed a musical instrument – the so-called 'vortex whistle' – where air, injected tangentially into a cylinder, swirls into a tube of smaller diameter. The sound thus emitted was found to have a discrete frequency proportional to flow rate. We recall that the frequency of the pure-tone noise in the similar swirling flow within the annular cascade was also proportional to the Mach number.

With the preceding background description in mind, we go right to the heart of the present theory: *acoustic streaming by the vortex whistle produces the Ranque–Hilsch effect.*

Here we redefine the vortex whistle in a broader sense as the pure-tone noise existing in any swirling flow, whether the fluid is contained within a single pipe or an annular duct; it is characterized by a spiralling waveform such as $\cos(m\theta + kz - \omega t)$, where θ and z are the circumferential and axial coordinates respectively, m and k are the corresponding wavenumbers and t is the time. ω is a *discrete* fundamental frequency determined uniquely for given values of m, k, swirl and the axial velocity.

The fact that the rotating fluid is endowed with a capacity to support such a spinning wave motion has long been established (e.g. Kelvin 1880; Greenspan 1968);

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† Syred & Beér (1972) also recorded regular sinusoidal fluctuations in the tube and suggested their possible connection with the energy separation.

like other pure-tone noise in flow, the vortex whistle, latent in swirling flow, emerges out of the selective amplification of background disturbances. This pure tone, we submit, brings the total-temperature separation into being.

Now this is not to suggest that the flow in the Ranque–Hilsch tube remains laminar; it is turbulent. But we shall argue and later verify experimentally (figure 5) that in the frequency spectrum, over and above turbulence, there soars a prominent peak of the vortex whistle possessing the discrete frequency ω. This organized disturbance dwarfs the random fluctuation, rendering the latter negligible by comparison. For a model problem of the Ranque–Hilsch tube posed as such, the frequency of the pure tone determined from the linearized inviscid solution will be found to increase proportionately to swirl. More important, ω, *thus calculated by neglecting the random components of disturbances, does agree closely with the measurement of the vortex whistle* (figure 6), with $m = 1$, the first tangential mode.

From this linearized treatment we proceed to the second-order analysis in order to calculate streaming; this is, of course, the temporal average of nonlinear terms or products between linear waves like $\cos^2(m\theta + kz - \omega t)$, whose time-dependent parts take the form $\cos(2m\theta + 2kz - 2\omega t)$ – an induced, spiralling wave with its frequency and wavenumbers twice those of the vortex whistle.

Concerning this *unsteady* part of the nonlinear terms, we confirm experimentally that such *a second harmonic precisely in the above form is conspicuously in evidence in the data obtained from the Ranque–Hilsch tube.* Its pre-eminence attests, and favours, the anticipated existence of its steady counterpart, or acoustic streaming, which the second harmonics must always keep in tow.

Concerning the *steady* part of the nonlinear terms or acoustic streaming, we shall obtain its expression near a cylindrical surface in general form (equation (22)) applicable to any swirl, and not limited to the one in a single pipe of the Ranque–Hilsch tube. We regard this as the centrepiece of the analytical results.

Applied to a model problem of the Ranque–Hilsch tube, where the base swirl initially imposed is assumed to be a Rankine vortex, the substitution of the calculated frequency ω of the vortex whistle into the above expression reveals the following: the tangential streaming near the tube periphery is always in the same direction as the steady swirl. Thus the streaming adds to the base swirl, and the total d.c. component of swirl near the tube periphery grows in magnitude. This is particularly so at lower tangential modes, since, as m decreases, the tangential streaming grows increasingly larger. This deforms the initial Rankine vortex, and a transition towards a forced vortex is now set in motion. The flow with the initially uniform total temperature becomes separated radially into a hotter gas near the tube periphery and a colder one near the centre. *The Ranque–Hilsch effect now takes shape.*

As mentioned, the vortex whistle in the Ranque–Hilsch tube turns out to be that corresponding to the first tangential mode, with $m = 1$. The pivotal point that this wave does cause the temperature separation, indicated by the theory, will be confirmed by the following test. On the Ranque–Hilsch tube, we install acoustic suppressors of the organ-pipe type (figure 7), tunable to the fundamental frequency ω of the vortex whistle, which corresponds to the first tangential mode, $m = 1$. *We thus attenuate the intensity of the vortex whistle at a tuned frequency, confirm that this induces the reduction of its second harmonic and, most important, verify that this does indeed reduce the total-temperature separation.* Figures 9–13 in § 5.2 display such results. For instance, in figure 9 the acoustic suppressor was tuned to 4·0 kHz; at the moment when the fundamental frequency of the vortex whistle reached the tuned frequency, its level plummeted abruptly by 25 dB, which in turn reduced its second harmonic by 14 dB. At this instant, the cold temperature at the tube centreline, which had gone down as low as −35 °C, immediately jumped to 0·6 °C – with a temperature rise equal to 35·6 °C. These results, more than anything else, seem to validate the heart of the present thesis. (Some readers may prefer to pass directly to § 5, which presents such results.)

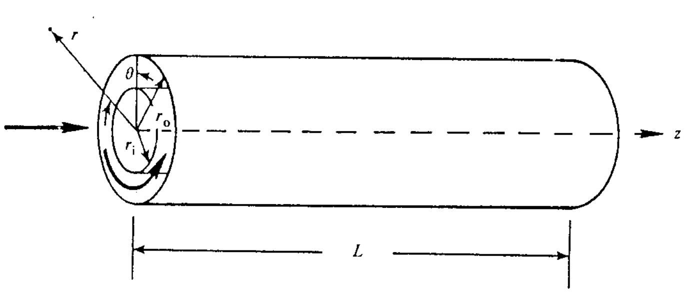

FIGURE 1. Definition sketch.

2. Statement of the problem

Although our present investigation centres on a single pipe, here we formulate, in anticipation of future work on co-axial cylinders, the following general problem: acoustic streaming in either a single pipe or an annular duct between two circular cylinders. The latter case is ske[illegible] figure 1, where r_o denotes the radius of the outer cylinder, r_i that of the inner cylinder. For a single pipe, corresponding to the Ranque–Hilsch tube of uniflow type, r_o still denotes the radius of the cylinder. The fluid is compressible and taken as a perfect gas. We assume that throughout the entire duct length L of interest, the two steady boundary layers formed over the cylinder surfaces are thin; in the inviscid annular region bounded and contained by them, both the circumferential and axial velocities are the predominant components of steady flow, as shown in figure 1. Superimposed upon this steady flow are the fully three-dimensional unsteady disturbances, whose streaming effects we are interested in.

Acoustic streaming is of course an induced steady or added d.c. component. As such, we have to distinguish it from the base steady flow initially imposed before the disturbances are set up. For brevity we henceforth refer to the latter simply as the steady part. Thus the flow at any point consists of three parts: the steady part, the a.c. components of the unsteady disturbances, and their d.c. components or streaming. The sum of the steady part and streaming is the total d.c. component.

For symbols, we use the following: (r, θ, z) are cylindrical coordinates with the corresponding velocity $\mathbf{q} = (u, v, w)$; p is the pressure, ρ the density, T the temperature, μ the coefficient of viscosity, $\nu = \mu/\rho$, Pr the Prandtl number, γ the ratio of specific heats; it turns out, to the order in which we are interested, that the second coefficient of viscosity does not appear. Both the specific heat c_p and Pr are taken, as usual, to be constant; the viscosity law is

$$\mu = \mu(T). \qquad (1)$$

The boundary condition on the cylindrical surfaces is $\mathbf{q} = 0$. The thermal condition depends, in general, on the details of unsteady heat transfer through a wall. For simplicity, we assume that either the wall temperature is maintained constant or the walls are insulated.

We first consider the steady part of the flow field. We assume that the steady flow is axisymmetric, and apply the standard outer and inner expansion (Van Dyke 1962*a*, *b*) in negative half-powers of the Reynolds number; we denote the leading term of the outer expansion or the inviscid flow by subscript 0.

As usual, the matching yields $u_0 = 0$ on the walls, and henceforth we are interested in the case where

$$u_0 = 0, \tag{2}$$

at any point of the outer or inviscid region. The rest of the leading terms of the outer expansion or inviscid flow are all taken to be functions of r only, independent of z (and of θ), in view of the assumption of thin boundary layers. Then the only non-trivial component of the equation of motion is the radial equilibrium

$$\rho_0 \frac{v_0^2}{r} = \frac{dp_0}{dr}. \tag{3}$$

For the energy equation, we assume that the leading term of the outer expansion of entropy remains constant everywhere.

The leading terms of the inner expansion of the steady part constitute of course the conventional compressible steady boundary-layer equations. On both cylindrical surfaces, the boundary layers are rotationally symmetric and develop in the z-direction. Their structures are complicated by the presence of two components of the inviscid stream: swirl and axial velocity. However, as shown later, insofar as we limit our attention to the leading term of acoustic streaming at large Reynolds number, the details of the steady boundary layer need not be worked out.

3. Small parameters and expansion series

We now turn to the unsteady disturbances, upon which our emphasis lies. In an effort to seek a tractable ingress to the problem, we assume that the a.c. components of the disturbances are organized, regular and endowed with a fundamental frequency ω; in comparison with them, we neglect the stochastic part of the disturbances according to the discussion of §1.

Now the only known analytical method of obtaining streaming is to resort to the use of an expansion scheme. To pave the way for the selection of the relevant small parameters, consider the linearized inviscid unsteady disturbances. Neglecting the second-order terms, from the set of equations (16*a*–*g*) in §4.1 we immediately recognize a fact of crucial importance: once a factor $\exp(im\theta + ikz - i\omega t)$ is separated from the amplitudes of the disturbances, the frequency ω does not appear by itself, but arises exclusively in the modified form of

$$-\omega + m\frac{v_0(r)}{r} + kw_0(r) = f(r). \tag{4}$$

This is only to be expected, because it embodies the Doppler shift caused by the steady inviscid swirl v_0 and the axial velocity w_0. Note that the shift is dependent upon the radial position.

In order to obtain acoustic streaming, these inviscid linear waves are to be imposed as external excitations upon the viscous flow near the walls; therefore, in forming a parameter containing a frequency, the relevant one is not the bare ω, but the above f evaluated at the cylindrical surface.

Henceforth, the leading term of the outer expansion of the steady flow, evaluated at R (the radius of the cylinder surface under study) is denoted by subscript e – the 'external' flow at the outer edge of the steady boundary layer; for instance

$$v_e = v_0(R), \quad w_e = w_0(R).$$

Then the appropriate frequency parameter relevant to the acoustic streaming near R is given by

$$\alpha = \tilde{v}_e / f_e R, \tag{5a}$$

where

$$f_e = -\omega + m\frac{v_e}{R} + kw_e, \tag{5b}$$

and $\tilde{v}_e$ is a measure of the *unsteady* disturbance, taken specifically as the amplitude of the linearized inviscid fluctuation (the 'external' unsteady flow) in the circumferential direction, evaluated at the outer edge of the *unsteady* boundary layer.

With respect to the lengthscale, in the problem at hand, three characteristic lengths distinguish themselves: the radius R of the cylinder, the unsteady-boundary-layer thickness $\delta_u = [\nu_e/|f_e|]^{\frac{1}{2}}$, and the steady-boundary-layer thickness $\delta = LRe^{-\frac{1}{2}}$, where Re is the Reynolds number based upon L. From the three lengths, we form two additional parameters β and ϵ, defined by

$$\beta \equiv \frac{\delta_u}{R} = \left[\frac{\nu_e}{|f_e|}\right]^{\frac{1}{2}} \frac{1}{R}, \tag{6}$$

$$\epsilon \equiv \frac{\delta}{\delta_u} = \frac{L}{\beta R\, Re^{\frac{1}{2}}}. \tag{7}$$

As regards δ_u and β, the absolute value of f_e is needed in the above; for, as seen from (5*b*), its sign can and does switch, depending upon the relative magnitudes of ω, v_e and w_e. Note that as the steady flow varies the unsteady-boundary-layer thickness δ_u can change, even for a given frequency. For example, δ_u may increase when f_e is decreased by an increase in v_e only. What this implies is, in effect, that *the unsteady boundary layer can become thicker than the steady boundary layer, particularly at high Reynolds number.*

We direct our interest to a perturbation expansion where

$$\alpha, \beta, \epsilon \ll 1. \tag{8}$$

Physically this may be visualized as follows. The amplitude of unsteady disturbances is small and, compared with the lengthscale of the cylinder radius, the *unsteady* boundary layer is thin. Within the unsteady layer an even thinner *steady* boundary layer is embedded.† Therefore, near any cylindrical surface the structure of the flow divides radially into three layers: (*a*) the steady boundary layer adherent to the wall, (*b*) the middle layer, which stretches from the outer edge of the steady boundary layer to that of the unsteady boundary layer, and (*c*) the inviscid core outside the unsteady boundary layer.

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† We assume that, for the steady and unsteady boundary layer alike, the thickness of the thermal boundary layer is of the same order as that of the momentum boundary layer, Pr being around unity.

With regard to the governing equations whose flow variables are to be expanded in power series of α, β and ϵ in order to obtain the streaming, one should, in principle, start with the full unsteady compressible Navier–Stokes equations rather than the boundary-layer equations. The reason is as follows.

As is well known, acoustic streaming stems from the temporal average of the products between the first-order quantities. Thus, if we start from the conventional boundary-layer equations, which correspond of course to the first-order approximation to the full Navier–Stokes equations, questions may arise about the effects of what is collectively called the higher-order approximation to boundary-layer theory (e.g. Van Dyke 1969) on streaming. In this connection, upon treating the problem of streaming around an oscillating cylinder, Stuart (1966) justly voiced a note of caution on the possible effect of curvature, which could be of the same second order as streaming itself. In the present case we are besieged by more than one effect of possible second-order correction. For example, both steady and unsteady boundary layers formed over the cylindrical surfaces present the problem of a steady as well as an unsteady displacement thickness; the fluctuation of temperature gives rise to changes in the viscosity which, coupled with a temporal variation in strains, might beget additional Reynolds stresses; the flow being compressible, even the effect of the second coefficient of viscosity may have to be assessed, as has in fact been done by Van Dyke (1962*a*) for steady compressible boundary layers.

To face these problems once and for all, the present author (Kurosaka 1980) abandoned the standard boundary-layer equations, and started afresh from the full compressible and unsteady Navier–Stokes equations, retaining even the second coefficient of viscosity. Under the conditions of (8), the matched asymptotic expansion in α, β and ϵ was used to ferret out the leading term of acoustic streaming, which was shown to be $O(\alpha)$; in the middle layer, the radial coordinate was scaled to δ_u; and the steady flow fields, corresponding to those outside the steady boundary layer, were expanded around the values at its outer edge, in a manner analogous to the procedure of Cole (1968, p. 16). Then, in the middle-layer equations up to the order of α, *the steady-flow properties appear as constant* evaluated at the outer edge of the steady boundary layer, rather than as the spatially changing variables, the reason being that δ_u *is much smaller than the scale of the inviscid core, the cylinder radius.* As regards the region within the steady boundary layer, whose complexity has been pointed out in §2, the difficulty was circumvented *by transferring* the boundary condition on the unsteady flow from the wall *to the innermost edge of the middle layer*; this in effect bypasses the steady boundary layer – a procedure wholly justifiable for $O(\alpha)$, not to mention $O(1)$ terms, because the ratio of the thickness of the steady boundary layer to R is $O(\epsilon\beta)$.

The consequences of the matched asymptotic expansion are as follows. As far as the terms up to $O(\alpha)$ are concerned, the governing equations in the middle layer are independent of such second-order effects as surface curvature, steady or unsteady displacement, or of the second coefficient of viscosity. (That the leading term of the streaming near the boundary remains unaffected by surface curvature and unsteady displacement is not unexpected from the analysis of Riley (1967), who confirmed this for an incompressible stationary-flow problem). In fact, the equations are none other than those derivable from the *conventional boundary-layer equations* – where the *base steady-state variables are evaluated at the outer edge of the steady boundary layer*, those with subscript e – and they are subject to the usual wall boundary conditions.

In the light of this justification and in an effort to avoid the inevitably elaborate algebra involved in starting from the full Navier–Stokes equations, we take the following standard boundary-layer equations as those governing the flow in the middle layer:

$$\frac{\partial\rho}{\partial t}+\frac{\partial(\rho u)}{\partial n}+\frac{1}{R}\frac{\partial(\rho v)}{\partial\theta}+\frac{\partial(\rho w)}{\partial z}=0, \tag{9a}$$

$$\frac{\partial p}{\partial n}=0, \tag{9b}$$

$$\rho L_{\mathrm{m}}(v)=-\frac{1}{R}\frac{\partial p}{\partial\theta}+\frac{\partial}{\partial n}\left(\mu\frac{\partial v}{\partial n}\right), \tag{9c}$$

$$\rho L_{\mathrm{m}}(w)=-\frac{\partial p}{\partial z}+\frac{\partial}{\partial n}\left(\mu\frac{\partial w}{\partial n}\right), \tag{9d}$$

$$\rho c_p L_{\mathrm{m}}(T)-L_{\mathrm{m}}(p)=\frac{c_p}{Pr}\frac{\partial}{\partial n}\left(\mu\frac{\partial T}{\partial n}\right)+\mu\left[\left(\frac{\partial v}{\partial n}\right)^2+\left(\frac{\partial w}{\partial n}\right)^2\right], \tag{9e}$$

$$p=c_v(\gamma-1)\rho T, \tag{9f}$$

where

$$L_{\mathrm{m}}=\frac{\partial}{\partial t}+u\frac{\partial}{\partial n}+\frac{v}{R}\frac{\partial}{\partial\theta}+w\frac{\partial}{\partial z}$$

and $n = r-R$ is the boundary-layer coordinate normal to the surface.

In accordance with our preceding discussion and subject to the restrictions imposed by (8), we expand the flow variables around those steady values with subscript e, the external flow at the outer edge of the steady boundary layer, in the form

$$\left.\begin{aligned} u &= u'+u''+\ldots,\\ v &= v_{\mathrm{e}}+v'+v''+\ldots,\\ w &= w_{\mathrm{e}}+w'+w''+\ldots,\end{aligned}\right\} \tag{10}$$

and so forth, where v_{e}, w_{e}, ... are constants. The rest of the terms represents the unsteady disturbances; the single-primed terms, when divided by the amplitude of disturbances $\tilde{v}_{\mathrm{e}}$, are $O(1)$; the d[illegible]imed terms, rendered in the same form, are $O(\alpha)$; for example

$$O\left(\frac{u'}{\tilde{v}_{\mathrm{e}}}\right)=1,\quad O\left(\frac{u''}{\tilde{v}_{\mathrm{e}}}\right)=\alpha. \tag{11}$$

The temporal averages of u'', v'' and w'' give the leading terms of the acoustic streaming; as mentioned, the terms corresponding to the $O(\beta)$, $O(\epsilon)$ and other higher orders do not contribute to the leading term of the d.c. component, and are neglected in the above expansion. In accordance with our aforementioned conclusions drawn by the method of the matched asymptotic expansions, the boundary conditions for the unsteady disturbances are given, at $n = 0$, as

$$\left.\begin{aligned} &u'=u''=v'=v''=w'=w''=0,\\ &T'=T''=0\quad\text{or}\quad\frac{\partial T'}{\partial n}=\frac{\partial T''}{\partial n}=0.\end{aligned}\right\} \tag{12}$$

The flow in the inviscid core is, of course, governed by the following inviscid equations:

$$\frac{\partial \rho}{\partial t}+\frac{1}{r}\frac{\partial(r\rho u)}{\partial r}+\frac{1}{r}\frac{\partial(\rho v)}{\partial \theta}+\frac{\partial(\rho w)}{\partial z}=0, \tag{13a}$$

$$\rho\left[L_1(u)-\frac{v^2}{r}\right]=-\frac{\partial p}{\partial r}, \tag{13b}$$

$$\rho\left[L_1(v)+\frac{uv}{r}\right]=-\frac{1}{r}\frac{\partial p}{\partial \theta}, \tag{13c}$$

$$\rho L_1(w)=-\frac{\partial p}{\partial z}, \tag{13d}$$

$$L_1\left(c_p \log\frac{T}{p^{(\gamma-1)/\gamma}}\right)=0, \tag{13e}$$

$$p=c_v(\gamma-1)\rho T, \tag{13f}$$

where

$$L_1=\frac{\partial}{\partial t}+u\frac{\partial}{\partial r}+\frac{v}{r}\frac{\partial}{\partial \theta}+w\frac{\partial}{\partial z}.$$

The corresponding expansions for the flow variables in the inviscid core are given by

$$\left.\begin{aligned} u &= u'+u''+\ldots,\\ v &= v_0(r)+v'+v''+\ldots,\\ w &= w_0(r)+w'+w''+\ldots, \end{aligned}\right\} \tag{14}$$

and the like, where the orders of the primed terms are the same as those in the middle layer, (11).

4. First- and second-order problems; streaming

Substitution of the expansion series from §3 into the corresponding governing equations yields a set of equations appropriate for each order. In this section we assemble them in hierarchic order and present solutions, together with some comments on salient features.

4.1. *First order in inviscid core*

We set

$$u'=\tilde{u}(r)\,e^{i(m\theta+kz-\omega t)},\quad v'=\tilde{v}(r)\,e^{i(m\theta+kz-\omega t)}, \tag{15}$$

and so forth, and obtain the following set of linearized equations in the inviscid core:

$$i\left(f\tilde{\rho}+\frac{1}{r}\rho_0 m\tilde{v}+\rho_0 k\tilde{w}\right)+\frac{1}{r}\frac{d}{dr}(r\rho_0\tilde{u})=0, \tag{16a}$$

$$if\tilde{u}-2\frac{v_0}{r}\tilde{v}=\frac{\tilde{\rho}}{\rho_0^2}\frac{dp_0}{dr}-\frac{1}{\rho_0}\frac{d\tilde{p}}{dr}, \tag{16b}$$

$$if\tilde{v}+\left(\frac{dv_0}{dr}+\frac{v_0}{r}\right)\tilde{u}=-\frac{im}{r\rho_0}\tilde{p}, \tag{16c}$$

$$if\tilde{w}=-\frac{ik}{\rho_0}\tilde{p}, \tag{16d}$$

$$\frac{\tilde{T}}{T_0}-\frac{\gamma-1}{\gamma}\frac{\tilde{p}}{p_0}=0, \tag{16e}$$

$$\tilde{p}=c_v(\gamma-1)(\rho_0\tilde{T}+T_0\tilde{\rho}), \tag{16f}$$

subject to the boundary condition

$$\tilde{u}(r=R)=0\quad (R=r_1,\,r_0). \tag{16g}$$

In deriving them, use is made of (3), and w_0 is assumed to be constant.

Note that in the above the frequency ω appears solely in the form of f, defined by (4), as mentioned previously; once the profile of the steady swirl $v_0(r)$ is specified, (16a–g) are complete and determine ω as a function of m, k and cylinder geometry. Such a frequency–swirl relationship, obtained by both an analytical and numerical method, is presented in the appendix for the specific swirl distribution corresponding to the Rankine vortex with a single pipe.

Inspection of (16) also reveals that if $\tilde{u}$ is imaginary, the other amplitudes, $\tilde{v}$, $\tilde{w}$, $\tilde{\rho}$ etc., are real: the real parts of u' and of all the others oscillate with a phase difference equal to $\frac{1}{2}\pi$. This implies that when we proceed to the second-order terms the temporal averages of the cross-products between u' and any other flow variable must vanish.

4.2. *First order in middle layer*

We set

$$u'=\tilde{u}(n)\,e^{i(m\theta+kz-\omega t)},\quad v'=\tilde{v}(n)\,e^{i(m\theta+kz-\omega t)}, \tag{17}$$

and so forth, and obtain the following linearized viscous equations in the middle layer:

$$i\left(f_c\tilde{\rho}+\frac{1}{R}\rho_c m\tilde{v}+\rho_c k\tilde{w}\right)+\rho_c\frac{d\tilde{u}}{dn}=0, \tag{18a}$$

$$\tilde{p}=-\frac{R}{m}\rho_c f_c\tilde{v}_c, \tag{18b}$$

$$i\rho_c f_c\tilde{v}=i\rho_c f_c\tilde{v}_c+\mu_c\frac{d^2\tilde{v}}{dn^2}, \tag{18c}$$

$$i\rho_c f_c\tilde{w}=i\frac{kR}{m}\rho_c f_c\tilde{v}_0+\mu_c\frac{d^2\tilde{w}}{dn^2}, \tag{18d}$$

$$ic_p\rho_c f_c\tilde{T}+i\frac{\rho_c R}{m}f_c^2\tilde{v}_c=\frac{c_p}{Pr}\mu_c\frac{d^2\tilde{T}}{dn^2}, \tag{18e}$$

$$-\frac{R\rho_c}{m p_c}f_c\tilde{v}_c=\frac{\tilde{T}}{T_c}+\frac{\tilde{\rho}}{\rho_c}, \tag{18f}$$

where $\tilde{v}_c$, as mentioned, is the amplitude of v' in the inviscid core, evaluated on the cylinder surface; the boundary conditions are

$$\tilde{u}=\tilde{v}=\tilde{w}=0,\quad \tilde{T}=0\quad\text{or}\quad\frac{d\tilde{T}}{dn}=0\quad (n=0), \tag{18g}$$

$$\tilde{v}=\tilde{v}_c,\quad \tilde{w}=\frac{kR}{m}\tilde{v}_c,\quad \tilde{T}=-\frac{R}{c_p m}f_c\tilde{v}_c,\quad \tilde{\rho}=-\frac{R\rho_c}{m a_c^2}f_c\tilde{v}_c\quad (|n|\rightarrow\infty), \tag{18h}$$

where (18b) expresses the constancy of the pressure across the middle layer, and (18h) the matching of flow variables at its outer edge, where all are expressed in terms of $\tilde{v}_e$. $|n| \to \infty$ is to be interpreted in the usual matching sense, with the radial coordinate scaled to δ_u; $a_e^2 = c_v\gamma(\gamma-1)T_e$.

Given $\tilde{v}_e$, (18) is completely determined; the solution corresponding to a thermal condition on the wall, $\tilde{T} = 0$, is given by

$$\tilde{u} = \tilde{v}_e \frac{im}{R}\operatorname{sgn}(n)\left\{|n|\left[\left(\frac{Rf_e}{ma_e}\right)^2 - 1 - \left(\frac{kR}{m}\right)^2\right] + \left[1+\left(\frac{kR}{m}\right)^2\right]\frac{1}{\Delta}g + \left(\frac{Rf_e}{ma_e}\right)^2\frac{\gamma-1}{Pr^{\frac{1}{2}}\Delta}(1-h)\right\}, \tag{19a}$$

$$\tilde{v} = \tilde{v}_e g, \quad \tilde{w} = \tilde{v}_e\frac{kR}{m}g, \tag{19b, c}$$

$$\tilde{\rho} = -\tilde{v}_e\frac{R\rho_e f_e}{ma_e^2}[1+(\gamma-1)h], \tag{19d}$$

$$\tilde{p} = -\tilde{v}_e\frac{R\rho_e}{m}f_e, \quad \tilde{T} = -\tilde{v}_e\frac{Rf_e}{c_p m}(1-h), \tag{19e, f}$$

where

$$g = 1-\exp(-\Delta|n|), \quad h = \exp(-Pr^{\frac{1}{2}}\Delta|n|),$$

$$\operatorname{sgn}(n) = \begin{cases} 1 & (n>0), \\ -1 & (n<0), \end{cases}$$

$$\Delta = \sqrt{\tfrac{1}{2}}(1\pm i)\left|\frac{f_e}{\nu_e}\right|^{\frac{1}{2}} \quad (+ \text{ sign for } f_e > 0; \quad - \text{ sign for } f_e < 0).$$

For a thermally insulated wall, the results can be obtained formally from (19) by letting $Pr \to \infty$. With regard to the fluctuating component of viscosity μ' needed for the next second-order calculation, it is related to T' by the viscosity law

$$\mu' = \left.\frac{d\mu}{dT}\right|_{T=T_e} T'. \tag{19g}$$

4.3. *Second order in middle layer*

We set

$$u'' = \overline{u''} + \mathscr{R}[A(n)\,e^{2i(m\theta+kz-\omega t)}], \tag{20a}$$

$$v'' = \overline{v''} + \mathscr{R}[B(n)\,e^{2i(m\theta+kz-\omega t)}], \tag{20b}$$

and the like, where the first terms with overbars represent the temporal averages taken over a period of oscillation $2\pi/\omega$; $\mathscr{R}$ denotes the real part. The leading terms of acoustic streaming correspond to these time averages, or d.c. components, which are functions of r only, independent of θ and z. The second terms of (20a, b) correspond to the induced second harmonics, $A(n)$ and $B(n)$ being their amplitudes.

In the corresponding governing equations we separate the d.c. components from the a.c. components, our interest being focused on the former. We first list those corresponding to the continuity, tangential and axial momentum equations and the equation of energy:

$$\rho_e\frac{d\overline{u''}}{dn} + \frac{d\overline{\rho' u'}}{dn} = 0, \tag{21a}$$

$$\rho_e\frac{d\overline{u'v'}}{dn} = \frac{d}{dn}\left(\overline{\mu'\frac{\partial v'}{\partial n}}\right) + \mu_e\frac{d^2\overline{v''}}{dn^2}, \tag{21b}$$

$$\rho_e\frac{d\overline{u'w'}}{dn} = \frac{d}{dn}\left(\overline{\mu'\frac{\partial w'}{\partial n}}\right) + \mu_e\frac{d^2\overline{w''}}{dn^2}, \tag{21c}$$

$$\begin{aligned} c_p\left[\overline{\rho'\frac{\partial T'}{\partial t}} + \rho_e\overline{u'\frac{\partial T'}{\partial n}} + \frac{\rho_e}{R}\overline{v'\frac{\partial T'}{\partial \theta}} + \frac{v_e}{R}\overline{\rho'\frac{\partial T'}{\partial \theta}} + \rho_e\overline{w'\frac{\partial T'}{\partial z}} + w_e\overline{\rho'\frac{\partial T'}{\partial z}}\right] \\ - \left[\frac{1}{R}\overline{v'\frac{\partial p'}{\partial \theta}} + \overline{w'\frac{\partial p'}{\partial z}}\right] \\ = \frac{c_p}{Pr}\frac{d}{dn}\left(\overline{\mu'\frac{\partial T'}{\partial n}}\right) + \frac{c_p}{Pr}\mu_e\frac{d^2\overline{T''}}{dn^2} + \mu_e\left[\overline{\left(\frac{\partial v'}{\partial n}\right)} + \overline{\left(\frac{\partial w'}{\partial n}\right)^2}\right]. \end{aligned} \tag{21d}$$

The boundary conditions are as follows:

$$\overline{u''} = \overline{v''} = \overline{w''} = 0, \quad \overline{T''} = 0 \quad \text{or} \quad \frac{d\overline{T''}}{dn} = 0 \quad (n=0), \tag{21e}$$

$$\lim_{|n|\to\infty}\frac{d\overline{v''}}{dn} = \lim_{|n|\to\infty}\frac{d\overline{w''}}{dn} = \lim_{|n|\to\infty}\frac{d\overline{T''}}{dn} = 0 \quad (n\to\infty). \tag{21f}$$

As for the boundary condition for $|n| \to \infty$, the original matching condition for $\overline{v''}$ reads

$$\overline{v''_{\mathrm{ml}}}(|n|\to\infty) = \overline{v''_{\mathrm{ic}}}(r=R),$$

where ml stands for the middle layer, ic for the inviscid core. The determination of streaming in the core will need this relationship; for the present middle layer, as in the usual case (Riley 1967; Batchelor 1967, p. 360), a weaker condition suffices: $\overline{v''_{\mathrm{ml}}}(|n|\to\infty)$ is finite, or equivalently the above $d\overline{v''}/dn \to 0$. Similar conditions apply to $\overline{w''}$ and $\overline{T''}$; one does not need conditions for $\overline{u''}$ at $|n| \to \infty$.

From (21), $\overline{u''}$, $\overline{v''}$, $\overline{w''}$ and $\overline{T''}$ are completely and uniquely determined in the middle layer. These d.c. components of velocity and temperature are dependent upon only the first-order terms in the middle layer, and independent of the streaming in the inviscid core; on the contrary, the latter will be dependent upon the former, regardless of the magnitude of the streaming Reynolds number R_s.

The solution of (21b) yields $\overline{v''}$. Corresponding to the thermal boundary condition where the fluctuation of temperature on the wall is maintained to be zero, the tangential streaming at the outer edge of the middle layer is given by

$$\begin{aligned} \overline{v''}(|n|\to\infty) = -\frac{m\tilde{v}_e^2}{2Rf_e}\left\{\frac{3}{2}\left[1+\left(\frac{kR}{m}\right)^2\right] - G + (\gamma-1)\frac{G}{Pr(Pr+1)}\right. \\ \left. - G(\gamma-1)\frac{Pr}{Pr+1}\left(\frac{3}{2}-\frac{T_e}{T_e+T_1}\right)\right\}, \end{aligned} \tag{22a}$$

where

$$G = (Rf_e/ma_e)^2. \tag{22b}$$

This we regard as one of our central results of analysis.

In obtaining this, an explicit relationship between viscosity and temperature appearing in (19g) is needed, and for this we have chosen Sutherland's formula

$$\mu = \mu^*\left(\frac{T}{T^*}\right)^{\frac{3}{2}}\frac{T^*+T_1}{T+T_1},$$

when μ^* is the viscosity at the reference temperature T^* and T_1 is, for air, 114 K.

Once the frequency–swirl relationship – obtainable from the linearized equation in the inviscid core for a specified swirl distribution – is substituted in f_e appearing in the above and defined by (5b), streaming is completely determined; we note that $\overline{v''}/\tilde{v}_e$ is $O(\alpha)$.

The axial streaming is related to the tangential streaming by

$$\overline{w''}(|n| \to \infty) = \frac{kR}{m}\overline{v''}(|n| \to \infty). \tag{23}$$

The streaming in the tangential and axial directions alike is independent of the actual value of viscosity, as expected. For our purposes we need not correct for the Stokes drift. We also emphasize that the expressions, as they stand, are not restricted to any specific radial profile of the steady flow field.

The radial streaming is given as

$$\overline{u''}(|n| \to \infty) = \pm \frac{\nu_e^{\frac{1}{2}}|f_e|^{\frac{1}{2}}}{2\sqrt{2}\,a_e^2}\tilde{v}_e^2\left\{1+\left(\frac{kR}{m}\right)^2+\left(\frac{Rf_e}{ma_e}\right)^2\frac{\gamma-1}{Pr^{\frac{1}{2}}}\right\}, \tag{24}$$

where + corresponds to the streaming near the surface $r = r_1$ and − to $r = r_0$; the radial streaming is now dependent upon the viscosity.

For a thermally insulated wall, the results can be obtained formally from the above by letting $Pr \to \infty$: for example, the tangential streaming becomes

$$\overline{v''}(|n| \to \infty) = -\frac{m\tilde{v}_e^2}{2Rf_e}\left\{\frac{3}{2}\left[1+\left(\frac{kR}{m}\right)^2\right]-G-G(\gamma-1)\left(\frac{3}{2}-\frac{T_e}{T_e+T_1}\right)\right\}, \tag{25}$$

and the like.

We now pause and confirm that the expressions (22) or (25) for the tangential streaming embrace the well-known results as their special cases. In the present representation of disturbances $e^{i(m\theta+kz-\omega t)}$, let $k = 0$, and in the place of θ introduce the circumferential distance x measured along the perimeter of the cylinder. Thus the disturbance may be regarded as a plane wave travelling in the x-direction: $e^{i(lx-\omega t)}$, where $l = m/R$. Furthermore, we restrict our attention to a fluid otherwise in a state of rest: $v_e = w_e = 0$. If the fluid is incompressible, G vanishes in the limit of $a_e \to \infty$ and (22) or (25) becomes

$$\overline{v''}(|n| \to \infty) = \frac{3}{4}\frac{l\tilde{v}_e^2}{\omega}. \tag{26}$$

This is but a classical result of streaming at the outer edge of the unsteady shear layer (e.g. Batchelor 1967, p. 360).

On the other hand, if the fluid is compressible, G is easily shown to be unity, and (25), corresponding to a thermally insulated wall, is reduced to

$$\overline{v''}(|n| \to \infty) = \frac{l\tilde{v}_e^2}{4\omega}\left\{1-(\gamma-1)\left(3-\frac{2T_e}{T_e+T_1}\right)\right\}. \tag{27}$$

The second term in the curly bracket corresponds to the streaming induced by the Reynolds stress that is caused by the variation of the viscosity due to the temperature fluctuation, the first term on the right-hand side of (21b). Numerically, this is by no means small; for instance, at $T_e = 20$ °C $= 293$ K, the second term of (27) is equal to 0·62, compared with unity of the first term. This notwithstanding, if we choose to ignore it, (27) becomes

$$\overline{v''}(|n| \to \infty) = \frac{l\tilde{v}_e^2}{4\omega}, \tag{28}$$

which is equal to $\tilde{v}_e^2/4a_e$, also a classical result for compressible flow (e.g. Lighthill 1978b, p. 347).

Compare (28), the streaming induced by a one-dimensional wave in the absence of base flow, with our general result (22) or (25) for the cylindrical waves where the base steady flow is present. We note that the factors outside the curly bracket of (22) or (25) can immediately be obtained from (28) by replacing l by m/R, and ω by f_e. The former corresponds, as noted, to the curvature effect. The latter is the Doppler effect apparent to an observer situated on the frame of reference in helical motion, advancing axially with w_e, while rotating w[illegible].

This much settled, we now turn to the radial momentum equation and the equation of state, related to the temporal averages of the pressure $\overline{p''}$ and density $\overline{\rho''}$. In contrast to the d.c. components of velocity and temperature, they cannot be determined from the following corresponding equations of the middle layer alone:

$$d\overline{p''}/dn = 0, \tag{29a}$$

$$\overline{p''} = c_v(\gamma-1)\left(\rho_e\overline{T''}+\overline{\rho'T'}+T_e\overline{\rho''}\right). \tag{29b}$$

If we were to solve them, $\overline{p''}$ and $\overline{\rho''}$ would have to be matched, at the outer edge of the middle layer, with the corresponding quantities in the inviscid core.

4.4. *Second order inviscid core*

Even in the core, we write the second-order disturbances in the form of (20); there, the temporal average term is still the leading term of the streaming in the core. Between the streaming in the middle layer and the inviscid core, the difference is that the latter is now driven by the former. This remains so regardless of the value of the streaming Reynolds number R_s, no matter how large or small (here R_s is defined as $R_s = (\alpha/\beta)^2$). However, as is now well known, the structure of the streaming outside the unsteady boundary layer is strongly dependent upon R_s, as first pointed out by Stuart (1963, p. 384).

If we take simply the temporal average of the second-order equations in the core, they, of course, correspond to the case of $R_s \ll 1$, and are given by

$$\overline{u''} = 0, \tag{30a}$$

$$-2\rho_0 v_0\frac{\overline{v''}}{r}-\frac{v_0^2}{r}\overline{\rho''}+\frac{1}{r}\frac{\partial}{\partial r}\left(r\rho_0\overline{u'^2}\right)-\frac{1}{r}\rho_0\overline{v'^2}-2\frac{v_0}{r}\overline{\rho'v'} = -\frac{\partial\overline{p''}}{\partial r}, \tag{30b}$$

$$\overline{p''} = c_v(\gamma-1)\left(\rho_0\overline{T''}+\overline{\rho'T'}+T_0\overline{\rho''}\right), \tag{30c}$$

with the boundary conditions $\overline{u''} = 0$ on the walls, and

$$\overline{v''_{\mathrm{ic}}}(r = R) = \overline{v''_{\mathrm{ml}}}(|n| \to \infty), \tag{30d}$$

$$\overline{w''_{\mathrm{ic}}}(r = R) = \overline{w''_{\mathrm{ml}}}(|n| \to \infty), \tag{30e}$$

$$\overline{T''_{\mathrm{ic}}}(r = R) = \overline{T''_{\mathrm{ml}}}(|n| \to \infty), \tag{30f}$$

with similar matching conditions for $\overline{p''}$ and $\overline{\rho''}$, as stated at the end of §4.3. In deriving them, we have taken due account of the phase relationship between u' and others, stated in §4.1. Missing from the above set is the energy equation which, once time-averaged, vanishes identically. The equation of continuity and the momentum equations in θ- and z-directions all collapse to yield (30a).

Since viscous terms are absent in (30a, b) the equations are of inviscid type. Equations (30d, e) correspond, however, to the boundary conditions on the 'moving

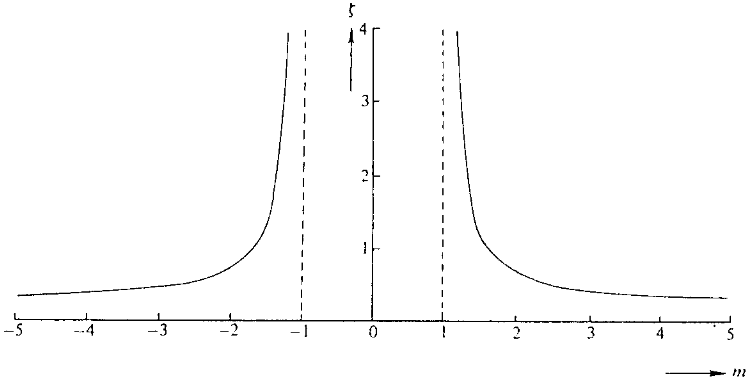

FIGURE 2. Tangential streaming near the tube periphery of a single pipe; $\lambda = 2, \kappa = 0$.

wall'; the fluid in the inviscid core is set in spiralling motion by the streaming in the middle layer. Equation (30f) corresponds to the prescribed surface temperature. In (30a–c), not only is the set of equations incomplete, but their depressed order ostensibly prevents them from satisfying the boundary conditions. However, this is in no way an indication of any kind of breakdown; the missing equations are to be supplied from the terms of *higher order* than the present second order. That this is so can best be illustrated by referring to the streaming induced by fluctuations in incompressible flow which is otherwise at rest. For $v_0 = 0$, we note that $\overline{v''}$ or the tangential velocity is now completely lost from (30b) or, for that matter, from everywhere else. It can be recovered only from the consideration of higher-order terms; in the terminology of the aforementioned matched asymptotic expansion of the present author, out of the viscous terms in the higher-order equations corresponding to $\tilde{v}_e O(\alpha\beta^2)$, we obtain

$$\nabla^4\overline{\Psi''} = 0, \quad \text{where} \quad \bar{v}'' = \frac{\partial\overline{\Psi''}}{\partial r},$$

which is the biharmonic equation originally due to Rayleigh (1884, p. 244) and derived anew by Riley (1967) using matched asymptotic expansions.

The foregoing discussion illustrates the fact that higher-order series terms in an expansion scheme are needed, in general, to determine the leading term of streaming in the core; this presents the entanglement of a formidable backward linking between terms in the series, as opposed to more direct forward linking.

We now turn to the case of $R_s \gg 1$. For incompressible fluid in the absence of base steady flow, the fundamental equation governing the exterior streaming was established by Riley (1967); it corresponds to the full steady Navier–Stokes equation, where R_s replaces the Reynolds number; thus, in order to solve this problem, nothing short of numerical computations (Duck & Smith 1979; Haddon & Riley 1979) seems to be effective. In the present situation of a compressible fluid where the base steady flow exists, there appears no reason to expect that the governing equation would become simpler.

In the face of these complications, we shall make no attempt to derive streaming in the core, being content instead that the phenomena of interest can be explained, as seen shortly, by inspecting the change in behaviour of the streaming at the outer edge of the middle layer, whose lead the streaming in the core follows.

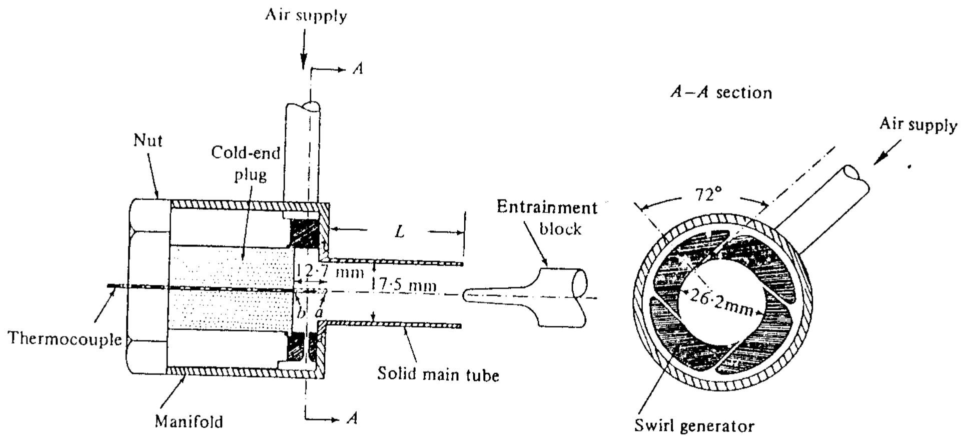

FIGURE 3. Layout of baseline configuration; solid main tube.

5. Application to Ranque–Hilsch tube

5.1. *Frequency and results of streaming analysis*

As a model 'inviscid' steady velocity profile corresponding to a Ranque–Hilsch tube of uniflow type, we consider a Rankine vortex within a single tube represented by

$$v_0(r) = \Omega r \quad (0 < r < r^*), \tag{31a}$$

$$v_0(r) = \Gamma/r \quad (r^* < r < r_0), \tag{31b}$$

where $\Gamma = \Omega r^{*2}$, and r^* denotes the radius at the interface between a forced and free vortex. We are interested in the unsteady disturbance with $k \rightarrow 0$ and the first radial mode.

As shown in the appendix, the frequency–swirl relationship for such a case, determined from linearized equations in the inviscid core, can be represented adequately by

$$\omega = \pm(|m| - 1 + \lambda^{-2|m|})\frac{\Gamma}{r^{*2}}, \tag{32}$$

± sign corresponding to $m > 0$ or $m < 0$ respectively. Although the relationship is derived by neglecting the second-order terms in the swirl, even at a swirl Mach number near unity, this analytical expression still compares favourably with the numerical results of Sozou & Swithenbank (1969) (see appendix). The linear dependence of the frequency on the swirl is a singularly distinguishing feature of the vortex whistle in a single pipe discovered first by Vonnegut (1954); see also Chanaud (1963, 1965).

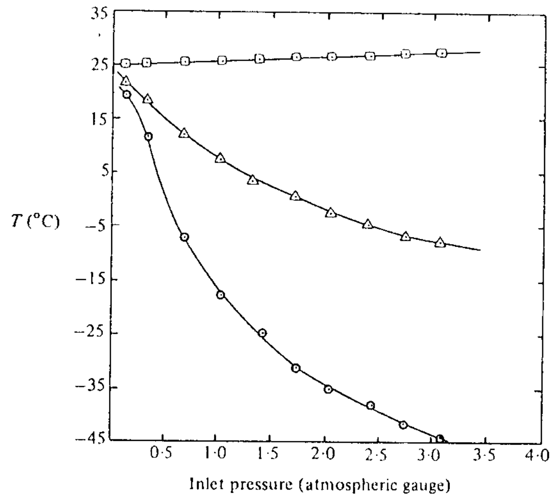

FIGURE 4. Temperature measurement for solid main tube; L = 4·19 cm: ⊡, inlet temperature; △, centreline temperature at position b of figure 3; ⊙, centreline temperature at position a of figure 3.

Substituting (32) into (22) or (25) and neglecting the second-order terms in the swirl, we obtain the streaming near the tube periphery:

$$\overline{v''}(|n| \to \infty) = \frac{\tilde{v}_c^2 \zeta}{\Gamma/r_0}, \tag{33a}$$

where

$$\zeta = \frac{3}{4}\frac{|m|}{(|m|-1+\lambda^{-2|m|})\lambda^2-|m|} \tag{33b}$$

and $\lambda = r_0/r^*$. This holds regardless of the thermal conditions, adopted in deriving (22) or (25), on the tube wall.

The above equation (33) shows that the streaming remains obviously the same even when m is replaced by $-m$, and furthermore it can readily be shown that its sign is always positive:

$$\overline{v''}(|n| \to \infty) > 0,$$

no matter what the values of λ and m may be. Positive values imply that the tangential streaming is in the same direction as the steady swirl; this in turn implies that streaming near the tube periphery augments the steady swirl. The magnitude of such streaming increases as the circumferential wavenumber m decreases. Figure 2 illustrates this, where ζ of (33b) is drawn as a continuous function of the circumferential wavenumber (though in reality, of course, it takes only integer values); λ is taken to be 2. Observe the increase of ζ as m is decreased; at $m = 1$, ζ grows infinitely large, whatever λ may be, as is easily seen from (33).

This added tangential velocity near the tube periphery converts the original Rankine vortex into a forced-vortex type. Even disregarding momentarily the effect of the static temperature gradient, this metamorphosis towards a forced vortex on its own tends to separate the total temperature: the colder stream near the centre and

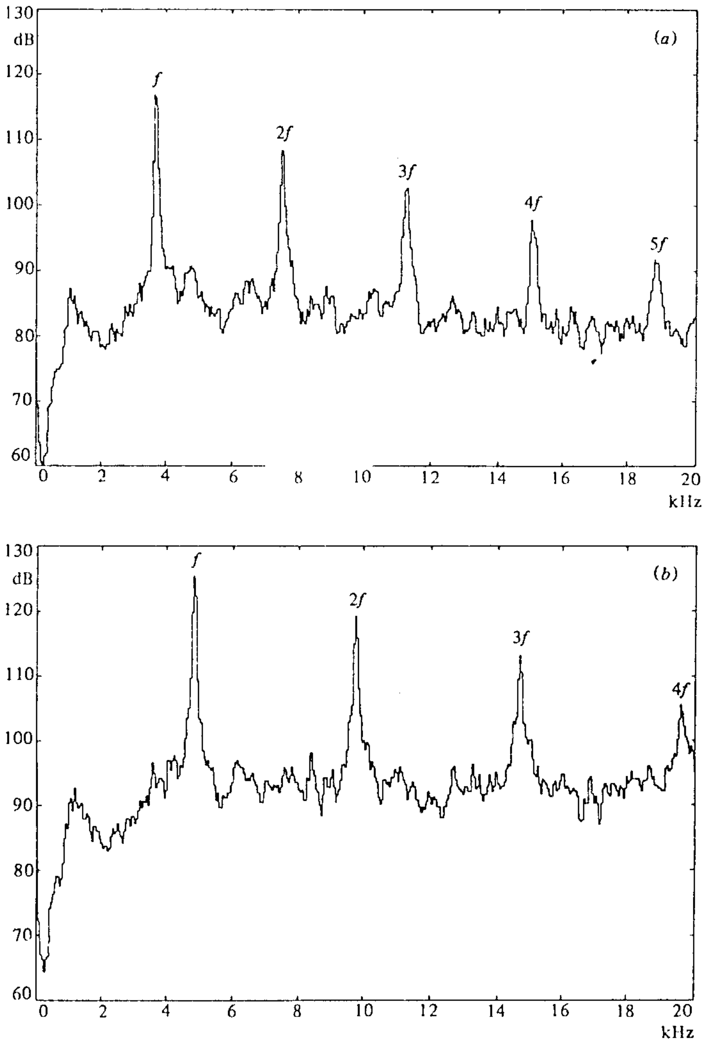

FIGURE 5. Frequency spectra for solid main tube; L = 4·19 cm: (a) inlet pressure = 1·02 (atmospheric gauge); (b) inlet pressure = 2·04 (atmospheric gauge).

the hotter stream near the periphery of the tube. This is precisely the onset of the Ranque–Hilsch effect.

As for the static temperature distribution, which in the inviscid core is determined by the total swirl, the static temperature for a forced vortex with sufficiently higher speed close to the periphery is lower than that of the Rankine vortex, provided that the temperature near the tube wall remains the same. This difference in static temperature further separates the total temperature.

Obviously, our analysis aims only at the onset of total-temperature separation. We

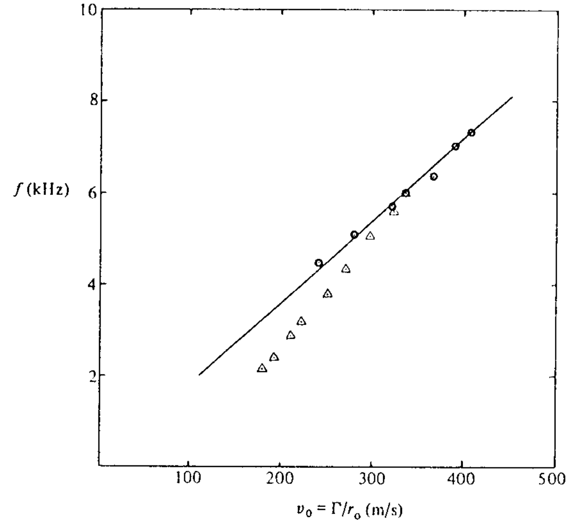

FIGURE 6. Fundamental frequency for solid main tube: ——, calculated; △, L = 4·19 cm; ⊙, L = 1·27 cm.

rest content to exhibit the initial conversion of swirl distribution, based upon the acoustic streaming, to a forced-vortex type and give no detailed calculation of the final equilibrium flow field.

Equation (33) further asserts that such a conversion of swirl should take place *regardless of the speed of the swirl*,† and regardless of the axial position within the tube, *even near its inlet*. This appears to fit in well with the previous observations: detailed measurements in the Ranque–Hilsch tube indeed show that a forced vortex is always found at any speed of swirl, and is formed immediately near the entrance to the tube even at a location as close as practically possible to the inlet nozzle (Hartnett & Eckert 1957; Scheller & Brown 1957; Lay 1959; Savino & Ragsdale 1961; Sibulkin 1962; Takahama 1965; Bruun 1969). The forced vortex occupies almost the entire cross-section, except of course for the neighbourhood of the boundary layer on the tube periphery. The radial separation of total temperature right at the tube entry, induced by the formation of the forced vortex, is also found by many experimenters. Contrast this with the ill-founded hypothesis of a gradual deterioration of a free vortex to a forced vortex in the direction of flow, which, although advocated as a mechanism of total temperature by Hilsch (1947) and Knoernschild (1948), has never been observed.

As stated, ζ becomes infinitely larger at $m = 1$, and obviously the analysis formally breaks down here. Despite this, we may expect that the trend of large streaming and the resulting total-temperature separation are indicative of what actually happens in a real flow. This we anticipate in analogy with the other similar resonance-like phenomena; in the spirit of the present investigation, we make no attempt to expend any elaborate computational effort on this, and instead we proceed to verify this directly by experiments described next.

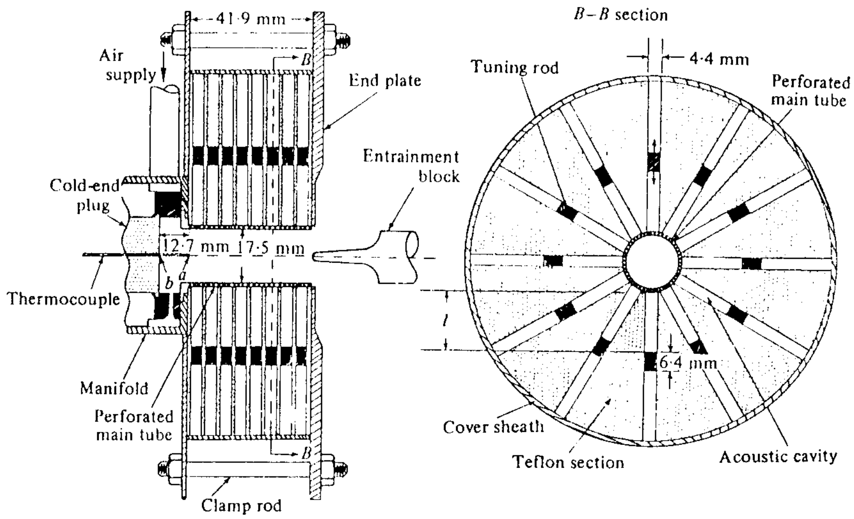

FIGURE 7. Layout of modified test rig; perforated main tube with acoustic suppressors.

5.2. *Experiments*

As a prototype of the test rig, a commercially available Ranque–Hilsch tube (Vortec Corporation, model no. 328–50) was acquired: in its original form a main pipe of 22·9 cm length and 1·75 cm diameter was connected to a manifold section, where a swirl generator, with four tangentially drilled slots inclined 72° from the radial direction, was housed. Pressurized air entered the manifold and separated into two counter-flows: the open end of the main pipe, where a flow-controlling globe valve was placed, served as the exhaust for the hotter air, while the colder air discharged from an opening placed at the other end of the manifold.

5.2.1. *Baseline configuration and measurements.* Subsequently, two modifications were added to the prototype. First, the cold end in the manifold was completely closed up by a Teflon plug; the air discharged only from the main pipe. This change to a uniflow arrangement was made to ease a direct comparison with the analytical model. Secondly, on the basis of the previous discussion, where we stated that the main pipe need not be long to create the total-temperature separation, the initial tube of L = 22·9 cm was replaced with a shorter brass tube of 4·19 cm. Figure 3 shows the entire layout of this uniflow test rig, which served as our baseline configuration.

With the length of the main pipe thus shortened, it was found that the use of the original globe valve at the end of the main pipe seriously affected the intensity of the pure-tone noise. Since the rig was eventually to be further modified so as to reduce the pure tone by the installation of acoustic suppressors, it was necessary to minimize such an acoustic interference in this baseline configuration. At the same time, however, a provision of some form of block at the end of the main pipe was required to prevent

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† Although the presence of the steady swirl in the denominator might convey the false impression that, as the swirl increases, the amount of streaming and hence the separation of the total temperature becomes less, what actually happens is that the amplitude of the disturbance $\tilde{v}_\theta$ grows faster, resulting in the net increase in $\overline{v''}$.

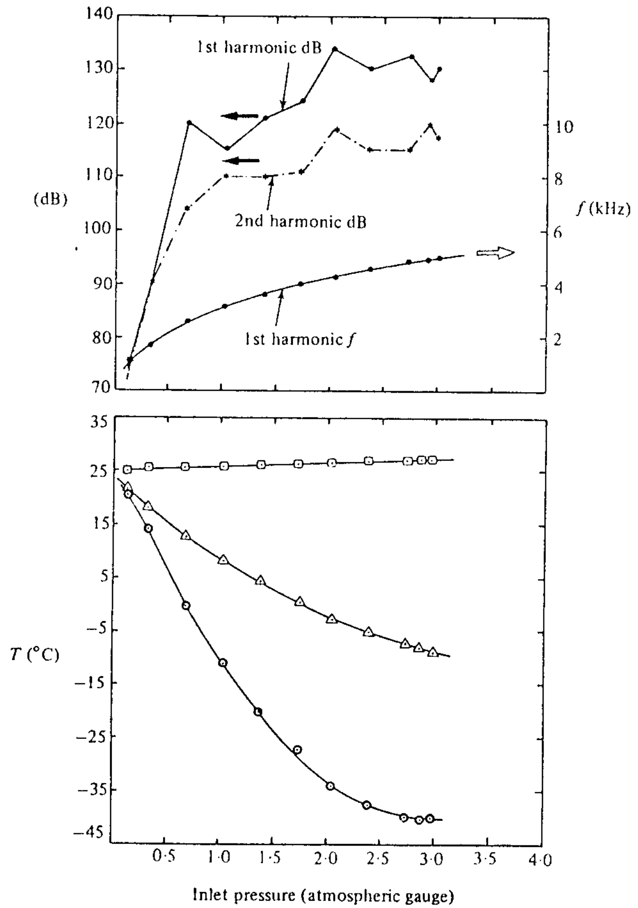

FIGURE 8. Temperature and acoustic measurement for perforated main tube with acoustic suppressors; l = 0 mm; ⊡, inlet temperature; △, centreline temperature at position b of figure 7; ⊙, centreline temperature at position a of figure 7.

the entrainment of the ambient air into the test section, which would otherwise have occurred owing to the lowered pressure at the core of the vortex. To meet these conflicting requirements, several conical shapes with different geometry were tried; among them the one with a contour shown in figure 3 was selected to block the entrainment. Unless otherwise stated, the tip of the block was always placed in the exit plane of the main pipe.

The pressurized air supplied from a separately located high-pressure tank entered the manifold via a pressure regulator, a flow-metering orifice and a sound muffler. The entire test assembly was housed inside a reverberation chamber.

A shielded type Chromel–Alumel thermocouple of 1·6 mm diameter (Omega SCASS-062G-6) was inserted through the cold-end plug. By moving it in the axial direction, the temperature at the centreline of the tube was traversed (see figure 3). A condenser microphone of 6·4 mm diameter (Bruel & Kjaer Type 4135), mounted nominally 91 cm away in the exit plane of the main pipe, monitored the sound; its output was fed into an FFT high-resolution signal analyser (Bruel & Kjaer Type 2033).

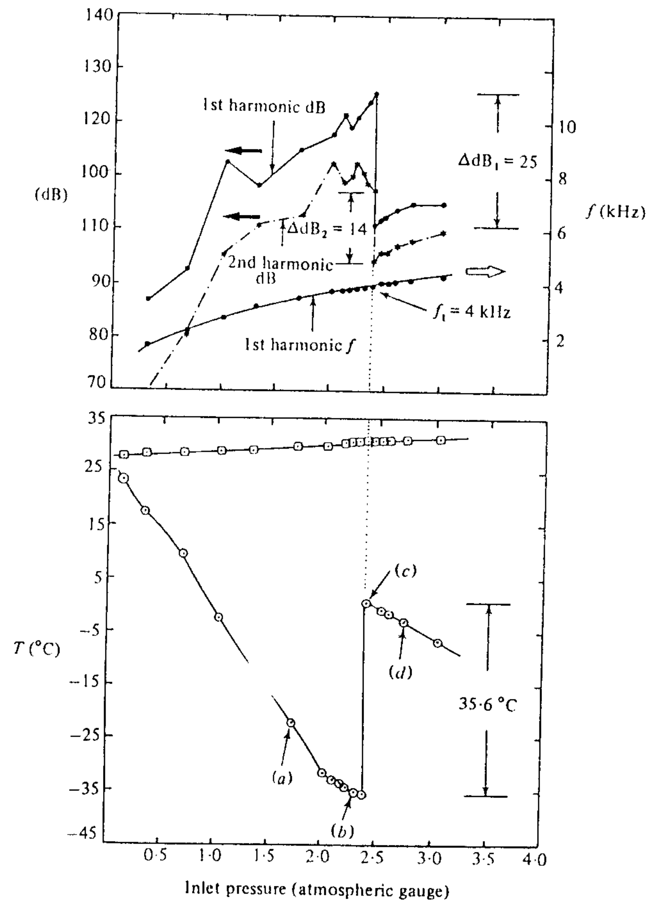

FIGURE 9. Temperature and acoustic measurement for perforated main tube with acoustic suppressors; l = 1·02 cm: ⊡, inlet temperature; ⊙, centreline temperature at position a of figure 7. (a), (b), (c) and (d) correspond to those of figure 10.

Figure 4 presents the dependence of the cold temperature at the centreline, together with the record of the inlet temperature. Two cold-temperature measurements are shown: one at the position where the thermocouple was flush with the cold-end plug (position b of figure 3) and the other at 12·7 mm away from it (position a). The latter corresponded to the coldest spot along the centreline. Observe that it went down as low as −45 °C. The presence of the coldest spot near the entrance to the main pipe agrees with the findings of others, referred to in §5.1; as stated,

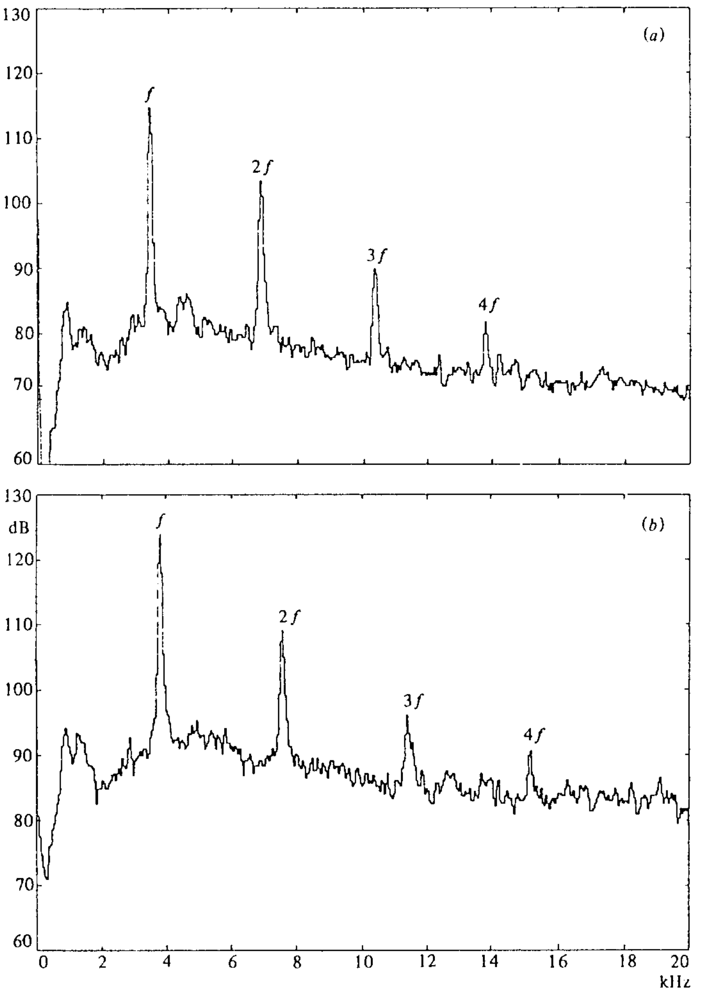

FIGURE 10 (a, b). For caption see facing page.

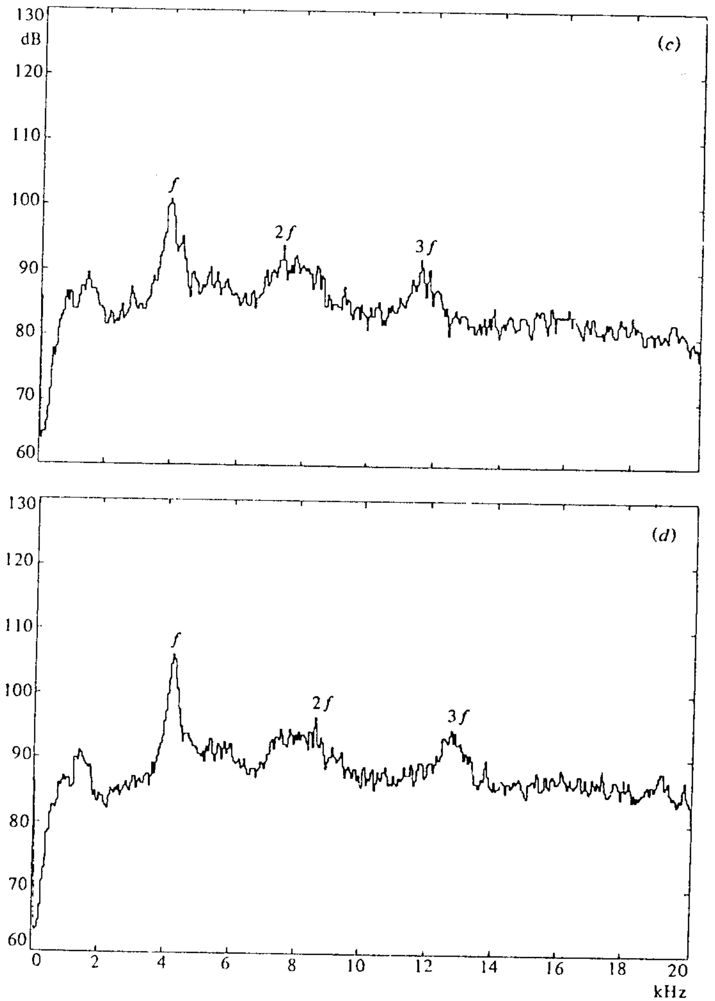

FIGURE 10. Frequency spectra for perforated main tube with acoustic suppressors; l = 1·02 cm: (a) inlet pressure = 1·70 (atmospheric gauge), (b) 2·31, (c) 2·38, (d) 2·72.

if the acoustic streaming is indeed a mechanism, it should cause the total-temperature separation even near the tube entrance. Note also in figure 4 that, as predicted, the temperature separation occurred even at low inlet pressure.

Figure 5 exemplifies the narrow-band frequency spectra, where we immediately recognize the predominant peaks of the first harmonic, with frequency $f = \omega/2\pi$ and its higher harmonics. The pure tones indeed overshadow the background noise, as postulated in § 1 and the analysis above. The two spectra of figure 5 at different inlet pressures illustrate, by comparison, the point that f increases as the inlet pressure is raised; note also the increase of the sound level at higher inlet pressure. A complete removal of the entrainment block from the main-pipe exhaust did not change f.

The tangential wavenumber m of the pure tones was determined by placing two microphones at different circumferential positions around the exhaust and in the exit plane of the main pipe and by measuring their phase difference ψ in the following ways: (i) observation of Lissajous figures displayed on a dual-beam oscilloscope, and (ii) direct measurement of ψ by a phase meter (Wavetek Model 750). By filtering only the signal corresponding to the fundamental frequency f, from two microphones placed 180° apart, ψ was found to be 180°; when they were 90° apart ψ = 90°. This means m = 1 for f; the pure tone with f corresponds to the first tangential mode of a spinning wave or the vortex whistle. The result is consistent with the measurements of Chanaud (1963) made for Vonnegut's 'vortex whistle'.

As for the second harmonic, by changing the filter frequency to $2f$, from two

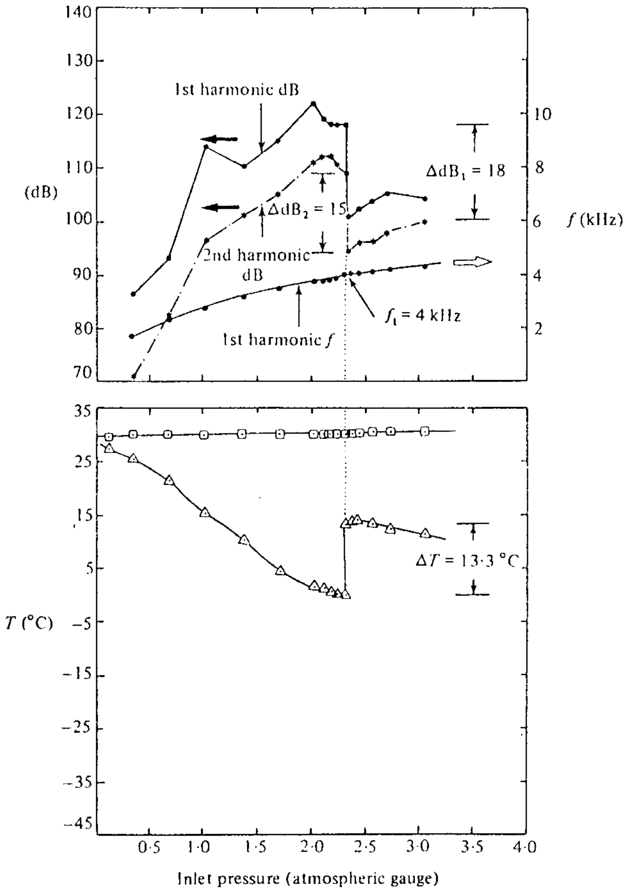

FIGURE 11. Temperature and acoustic measurements for perforated main tube with acoustic suppressors; $l = 1\cdot02$ cm: ☐, inlet temperature; △, centreline temperature at position b of figure 7.

microphones placed 90° apart, ψ was found to be 180°; for 45° apart $\psi = 90°$. Thus the wavenumber for $2f$ is $m = 2$. The doubling of the wavenumber when the frequency is doubled corresponds unmistakably to the induced harmonics as exemplified by the second terms of (20a, b), whose form is applicable to the inviscid core as well; this attests to the presence of the second-order disturbances, as expected from analysis. The wavenumbers of the third and fourth harmonics were identified, in a similar manner, to be equal to $m = 3$ and 4 respectively.

Concerning the first harmonic or the vortex whistle, by setting $m = 1$ (32) reduces to

$$f = \frac{\omega}{2\pi} = \frac{1}{2\pi}\frac{\Gamma}{r_0^2}. \tag{34}$$

This special formula appears to have first been obtained by Sozou & Swithenbank

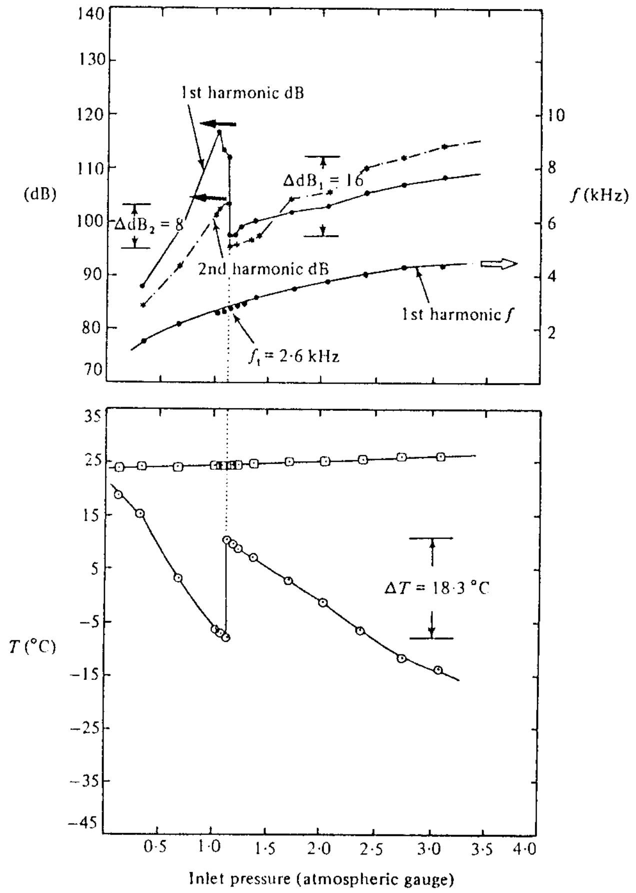

FIGURE 12. Temperature and acoust rement for perforated main tube with acoustic suppressors; $l = 1\cdot52$ cm: ☐, inlet temperature; ⊙, centreline temperature at position a of figure 7.

(1969), who apparently found it from inspection of their numerically computed results; see their figure 8(a). Given circulation Γ and the radius of the pipe r_0, we can predict from (34) the frequency† – fortuitously even without measuring r^*, the radius of the interface in a Rankine vortex. Γ may be estimated by assuming that its value around the tube periphery is equal to that around the circumference at the exit of the swirl generator. (Given the injection angle and injection-hole area, the tangential

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† Although the formula is derived for a Rankine vortex, it is also applicable to an actual forced vortex, which, on the evidence of the aforementioned measurements by previous investigators, is certainly formed within the present tube. The reason is that, because of the presence of the boundary layer near the tube periphery, any existent forced vortex is in effect a Rankine vortex with an enlarged region of a forced-vortex zone.

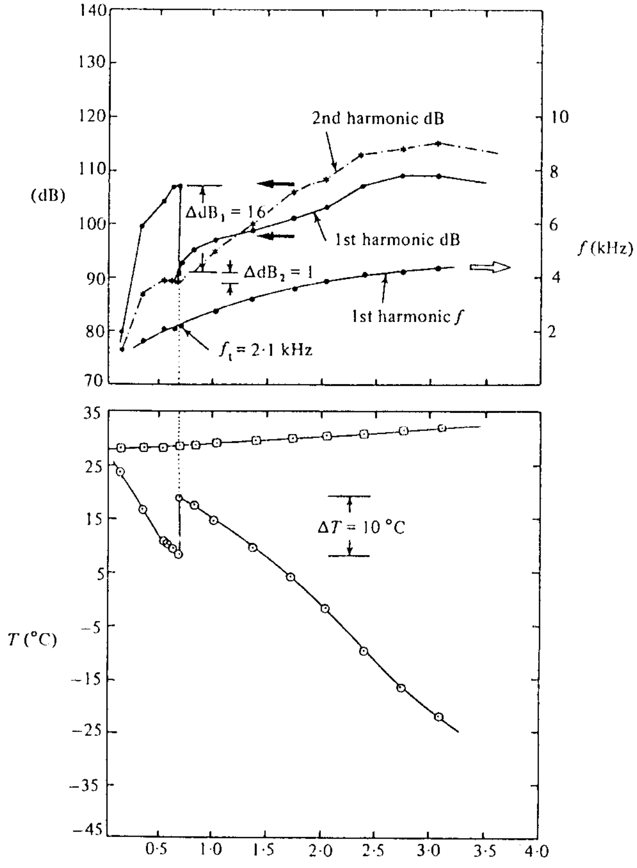

FIGURE 13. Temperature and acoustic measurement for perforated main tube with acoustic suppressors; $l = 2{\cdot}03$ cm: ⊡, inlet temperature; ⊙, centreline temperature at position a of figure 7.

velocity is estimated by the volume flow determined from the mass flow and a static pressure tap installed on the cold-end plug.) The frequency f thus calculated is shown in figure 6, together with the measured value corresponding to the tube length of 4·19 cm. Although the agreement may not be unreasonable, the obvious discrepancy is noticeable, and it was felt that this might be due to the slowdown of swirl through the pipe. Therefore a shorter version of the main tube $L = 1{\cdot}27$ cm was constructed and tested; although the frequency peaks were not as distinct as that for 4·19 cm, they were still well defined. The measured first harmonic, presented also in figure 6, now shows an excellent agreement with theory.

5.2.2. *Perforated main tube with acoustic suppressors.* Upon the completion of the baseline tests, the main tube of smooth brass was replaced with a perforated tube of the same diameter and length (figure 7); the tube was encased in an intermediate Teflon section, where acoustic cavities of 4·4 mm diameter, 8 in the axial and 12 in the circumferential direction, were drilled out. By inserting tuning rods of Plexiglas (6·4 mm long) into each cavity and varying its cavity length l the vortex whistle could be attenuated at any tuned frequency f_t. In order to reduce leakage, the intermediate section was covered by a sheath of Teflon and, furthermore, the entire test section was fastened by clamps.

With regard to the perforation on the main tube, holes of 1·2 mm in diameter were drilled out on the brass tube. In the initial arrangement, holes formed a staggered array, 45 rows in the circumferential direction and 25 in the axial direction. With the intermediate Teflon section fitted on the perforated tube, those holes on the tube that did not match with the acoustic cavities were identified and refilled with fibreglass filler in order to remove any unnecessary irregularity on the inner surface of the main tube. By this means, on the average a group of six holes upon the brass section matched with each acoustic cavity; over the main tube, the porosity became about 27 %.

These modifications, achieved after considerable trial and error, were found to suit our purpose: to reduce the vortex whistle as sharply as possible at the tuning frequency.

To ensure that the modification of the main tube from a smooth surface to a perforated one did not inadvertently alter the temperature separation, a test was run first with all cavity lengths set to zero. The results are shown in figure 8. Compared with the smooth tube, only the frequency was found to be noticeably affected, lowered by some 20 %, as might be expected from the slowdown of swirl due to the irregularity on the internal surface of the perforated tube. The cold temperature, measured at the coldest spot (position a of figure 7) was slightly higher for the perforated tube, and the sound level of the second harmonic slightly lower, but all in all they remained essentially unaffected; the peaks of the pure tones on the frequency spectra were as distinct and dominant as those for the smooth tube. Thus the perforated tube with zero acoustic cavity length still retained the characteristics of the smooth tube.

Next, the length of all the acoustic cavities was set to 1·02 cm and the thermocouple was left at the coldest spot. As the inlet pressure was slowly increased, the temperature at the cold spot kept on dropping, and at the same time both the frequency and intensity of the vortex whistle kept on increasing, as in the case of zero cavity – until, at $f = f_t = 4$ kHz, all of a sudden the sound became quieter, changing from a shrill whistle to a muffled hiss. At this instant, the cold temperature, which had gone down as low as −35 °C, immediately jumped up to 0·6 °C, with a temperature rise equal to 35·6 °C. The results are presented in figure 9, where, in addition to the temperature jump, the drops in the level of the first and second harmonics are shown. For the first harmonic, $\Delta dB_1 = 25$ dB, for the second harmonic $\Delta dB_2 = 14$ dB.

We display the successive change of frequency spectra around the present $f_t = 4{\cdot}0$ kHz in figure 10: (a) corresponds to $f = 3{\cdot}5$ kHz, below f_t, (b) $f = 3{\cdot}9$ kHz, just below f_t, (c) $f = f_t = 4{\cdot}0$ kHz, right at the tuned condition, and (d) $f = 4{\cdot}2$ kHz, above f_t. (The points corresponding to (a)–(d) appear in figure 9.) Comparison of the spectra clearly shows the following: the pure tones retained their spiky peaks right up to f_t (b); once $f = f_t$ (c), they suddenly became diminished and less distinct – 'spires' were lost. At the same time, the bases of the pure tones, as measured by the externally placed microphone, appear to become broadened ('haystacking'). Note also that between (b) and (c) the background noise level remained virtually the same.

As the frequency was increased further (*d*) the pure tone started to recover its spires. At the same time the cold temperature started to drop once again, as seen from figure 9.

We observe that at f_t the reduction of the sound level of the first harmonic, to whose frequency the cavity length was tuned, did cause the reduction in the second harmonic as well; this appears to add one more telling piece of evidence that the second harmonic is induced by the first harmonic. *Most importantly, the reduction of the first harmonic with* $m = 1$ *and its second harmonic did in fact reduce the temperature separation, as predicted by analysis.*

We cannot emphasize too strongly that the jump in temperature was not caused by any sudden change in the inlet temperature; the latter remained the same near f_t, before, after and in between, as is evident from the data shown in figure 9. Repeated tests confirmed that, once f approached f_t, just a slight increase in the upstream pressure precipitated the sudden reduction in noise level; or as was often the case, even when the upstream pressure was held constant, with f maintained equal to f_t, the precipitous drop in level took place spontaneously. Once the sound level became reduced and colder temperature shot up to a higher value, both stayed on that level until the upstream pressure was increased further.

Another phenomenon, compellingly evocative of the effect of acoustic streaming, could be easily perceived by sticking a finger near the exhaust: *below the tuning range, the flow discharging from the end of the main pipe was spiralling, outward and radially, over the end plate of figure* 7; *the region near the tube centreline was relatively quiescent – reminiscent of the eye of a hurricane. At the tuned condition, the swirling air suddenly rushed out along the tube centreline*; *the region over the end plate then became quieter.* This appears to bear vivid testimony to the expected *reversion* of the forced vortex to a Rankine vortex.

The data shown in figure 9 were taken with the cold-end thermocouple placed at the coldest spot (position *a* of figure 7). To confirm a point that the intrusion of the thermocouple did not inadvertently trigger the aforementioned sudden change, the thermocouple was withdrawn and placed flush with the cold-end plug (position *b*); then nothing lay in the way of the flow path. The results are shown in figure 11: at the same f_t as in figure 9 the sound level plummeted down again, accompanied by a temperature rise; the only major difference is the reduced amount of temperature jump, which corresponds to the higher temperature level at position *b*.

Figures 12 and 13 correspond to the cavity lengths of 1·52 and 2·03 cm, respectively; the cold-end thermocouple was placed at position *a* again. Other than the fact that f_t decreased as l increased, as expected, the basic features remained the same (the withdrawal of the thermocouple to position *a* again reproduced similar results).

Since the attenuation of the pure tone at f_t did not completely suppress the vortex whistle, as is evident from figure 10(*c*), by decreasing its intensity further it would appear possible to reduce the temperature separation further. In light of this – and although the temperature separation in the Ranque–Hilsch tube is not entirely attributable to the acoustic streaming induced by the vortex whistle, since the innermost core of any Rankine vortex must have total-temperature gradient caused by viscous stresses – it does not go too far to say that, for a vortex tube of uniflow type, the vortex whistle is in large part responsible for the Ranque–Hilsch effect.

An effort to obtain the radial distribution of temperature and pressure had to be aborted, since the radial immersion of a probe into flow was itself found to cause severe interference with the measurements; a similar difficulty was also reported by Reynolds (1962). In this connection, a similar, repeatedly observed point is worthy of note: the excitation of the vortex whistle to a level sufficient for a significant degree of temperature separation depends strongly on the internal geometry of the tube. For example, in an experiment with a solid, non-perforated tube, a thermocouple of 3·2 mm diameter was axially inserted from the exhaust to a point near the tube entrance; at that axial location, as the thermocouple was being moved radially towards the tube periphery, suddenly the vortex whistle vanished, resulting in the reduction of temperature separation again. When the thermocouple was either retracted towards the centre or withdrawn axially, the sound re-emerged, accompanied by the recovery of temperature separation. The impression gained by this and other similar incidents, which include the aforementioned difficulty of the radial immersion of the probe, is that the vortex whistle is always latent, but a certain subtle change in the internal tube geometry either excites it to a sufficient level or fails to do so. Although any exhaustive investigation into this is beyond the scope of the present investigation, the following fact is highly suggestive: in the commercially available Ranque–Hilsch tube the internal surface of the tube is slightly tapered to form a short diffuser section near the entrance to the main pipe (for the performance of this see Takahama & Yokozawa 1981).

6. Concluding remarks

In sum, departing radically from previous theories, we have demonstrated through an analysis and experiments that the acoustic streaming caused by the vortex whistle, a spinning wave with a discrete frequency in swirling flows, is in great part responsible for the Ranque–Hilsch effect in the vortex tube of uniflow type: near the periphery of the tube, the induced 'd.c.' tangential velocity adds to the steady swirl, transforming the latter to a forced vortex and leading to the total-temperature separation in the radial direction.

To the extent that the temperature separation arises owing to the unsteadiness in flow, the present subject is the converse of the phenomena of Rijke tubes and thermally driven acoustic oscillations of liquid helium (e.g. Clement & Gaffney 1960), where the difference in the temperature gives rise to unsteady disturbances. In a broader context, to the extent that we seek in the present mechanism an organized origin distinct from the stochastic process, this falls under the same morphological group as the study of the large-scale structure in a mixing layer (Brown & Roshko 1974).

The author is indebted to Fr C. E. Danforth for calling his attention to the phenomenon of the annular cascade and its connection with the Ranque–Hilsch effect; to Messrs J. R. Goodman, J. Q. Chu and R. D. Fizer for their invaluable and indefatigable assistance in carrying out the experiments; to Mr H. R. Bankhead for his encouragement; to Mr H. Hehmann for his advice on the suppressor design; to Drs M. E. Goldstein and W. H. Heiser and Mr F. R. Gliebe for helping him to clarify several points by raising pertinent questions; to referees and Drs Caruthers and Maus for providing valuable comments on the original manuscript; to many other investigators on related subjects, for answering his various queries about their work. The work is supported by the Air Force Office of Scientific Research under contract no. F49620-78-C-0045.

Appendix

In this appendix, we obtain from (16) the frequency ω corresponding to the steady swirl prescribed in § 5.

For the radial profile given by (31), the boundary conditions for the unsteady disturbances are

$$\tilde{u}(r=r_0)=0, \quad \tilde{p}, \tilde{u} \text{ continuous at } \quad r=r^*. \tag{A 1}$$

For a two-dimensional flow, Sozou & Swithenbank (1969) calculated the corresponding frequency relationship by a wholly numerical computation. (Although, as stated in § 4.1, the axial velocity is at present allowed to exist, as long as k is set equal to zero, w_0 drops out of (4) and the situation is the same as a two-dimensional flow.) Here we offer an approximate analytical representation, choosing $\tilde{p}$ as the primary dependent variable.

In the innermost core of the forced-vortex region defined by $0 < r < r^*$, we combine (16a–f) together and obtain the following single equation:

$$\frac{d^2\tilde{p}}{dr^2}+\left[\frac{1}{r}+\frac{(2\gamma-3)\,\Omega^2 r}{a_0^2(r)}\right]\frac{d\tilde{p}}{dr}+\left[\frac{2m\Omega^3}{\lambda_1 a_0^2(r)}+\frac{2(\gamma-2)\,\Omega^2}{a_0^2(r)}-\frac{m^2}{r^2}+\frac{D}{a_0^2(r)}\right]\tilde{p}=0, \tag{A 2a}$$

where $a_0(r)$, λ_1, and D are given by

$$a_0^2(r)=a_0^2(r=r_0)+\tfrac{1}{2}(\gamma-1)\left[\Omega^2(r^2-r^{*2})-\Gamma^2\left(\frac{1}{r^{2*}}-\frac{1}{r_0^2}\right)\right], \tag{A 2b}$$

$$\lambda_1=\Omega m-\omega, \tag{A 2c}$$

$$D=(\Omega m-\omega)^2-4\Omega^2, \tag{A 2d}$$

and $\tilde{u}$ is related to $\tilde{p}$ by

$$\tilde{u}=\frac{i}{\rho_0(r)\,D}\left\{\lambda_1 a_0^2(r)\frac{d\tilde{p}}{dr}+\tilde{p}\left[\frac{2ma_0^2(r)\,\Omega}{r}-(2-\gamma)\,\lambda_1 r\Omega^2\right]\right\}. \tag{A 2e}$$

On the other hand, in the free-vortex region defined by $r^* < r < r_0$, the governing equation is given by

$$\frac{d^2\tilde{p}}{dr^2}+\left[\frac{1}{r}+\frac{(2\gamma-3)\,\Gamma^2}{a_0^2(r)\,r^3}+\frac{4m\Gamma}{\lambda_2(r)\,r^3}\right]\frac{d\tilde{p}}{dr}$$
$$+\left[\frac{\lambda_2^2(r)}{a_0^2(r)}-\frac{m^2}{r^2}+\frac{2m(2\gamma-3)\,\Gamma^3}{\lambda_2(r)\,r^6 a_0^2(r)}+\frac{8(m\Gamma)^2}{\lambda_2^2(r)\,r^6}-\frac{4m\Gamma}{\lambda_2(r)\,r^4}+\frac{2(2-\gamma)\,\Gamma^2}{a_0^2(r)\,r^4}\right]\tilde{p}=0, \tag{A 3a}$$

where

$$a_0^2(r)=a_0^2(r=r_0)-\tfrac{1}{2}(\gamma-1)\,\Gamma^2\left(\frac{1}{r^2}-\frac{1}{r_0^2}\right), \tag{A 3b}$$

$$\lambda_2(r)=\frac{m\Gamma}{r^2}-\omega, \tag{A 3c}$$

and $\tilde{u}$ is related to $\tilde{p}$ by

$$\tilde{u}=\frac{i}{\rho_0(r)}\left\{\frac{a_0^2(r)}{\lambda_2(r)}\frac{d\tilde{p}}{dr}+\tilde{p}\left[\frac{2ma_0^2(r)\,\Gamma}{\lambda_2^2 r^3}-\frac{(2-\gamma)\,\Gamma^2}{\lambda_2 r^3}\right]\right\}. \tag{A 3d}$$

Equations (A 2a) and (A 3a) are the same as that obtained by Kerrebrock (1977).

We expand $\tilde{p}$, $\tilde{u}$ and ω in the power series of Ω:

$$\tilde{p}=\rho^{(0)}+\Omega\rho^{(1)}+\ldots, \tag{A 4a}$$

$$\tilde{u}=u^{(0)}+\Omega u^{(1)}+\ldots, \tag{A 4b}$$

$$\omega=\omega^{(0)}+\Omega\omega^{(1)}+\ldots. \tag{A 4c}$$

We confine our attention to $\omega^{(0)}=0$, which corresponds to the lowest eigenvalue for the case without swirl. We substitute (A 4a–c) into the aforementioned governing equations for $\tilde{p}$ and retain the terms up to $O(\Omega)$. This yields the following expression for $\omega^{(1)}$:

$$\omega^{(1)}=\pm(|m|-1+\lambda^{-2|m|}), \tag{A 5a}$$

where $\lambda=r_0/r^*$, and ω becomes from (A 4c)

$$\omega=\pm(|m|-1+\lambda^{-2|m|})\,\Omega, \tag{A 5b}$$

where the + sign corresponds to $m > 0$, and the − sign to $m < 0$.

Sozou & Swithenbank (1969) list the computed values ω in their table 3; in our table 1, we compare their results with our analytical formula (A 5b), the latter being shown in parentheses. The spaces left blank are unavailable from Sozou & Swithenbank's table. On the whole, our expression appears to be satisfactorily close to the numerical results, as long as the swirl Mach number, defined by them as $\Omega r^*/a_0(0)$, remains below 1.

$\frac{\Omega r^*}{a_0(0)}$	λ	$\omega r^*/a_0(0)\,m$				λ	$\omega r^*/a_0(0)\,m$			
		$m=1$		$m=2$			$m=1$		$m=2$	
0·10	1·09	0·0841	(0·0841)	0·0854	(0·0854)	1·03	0·0942	(0·0942)	0·0944	(0·0944)
0·10	1·5	0·0444	(0·0444)	0·0598	(0·0598)	1·2	0·0694	(0·0694)	0·0741	(0·0741)
0·10	5	0·0040	(0·0040)	0·0050	(0·0050)	2	0·0250	(0·0250)	0·0531	(0·0531)
0·25	1·09	0·2099	(0·2104)	0·2132	(0·2135)	1·03	0·2353	(0·2356)	0·2359	(0·2361)
0·25	1·5	0·1105	(0·1111)	0·1490	(0·1497)	1·2	0·1729	(0·1736)	0·1848	(0·1852)
0·25	5	0·0100	(0·0100)	0·1246	(0·1252)	2	0·0622	(0·0625)	0·1321	(0·1328)
0·50	5	0·0199	(0·0200)	—	—	2	0·1222	(0·1250)	0·2603	(0·2656)
0·75	5	0·0296	(0·0300)	—	—	2	0·1807	(0·1875)	0·3814	(0·3984)
1·0	5	0·0392	(0·0400)	—	—	2	0·2367	(0·2500)	0·4941	(0·5312)

TABLE 1

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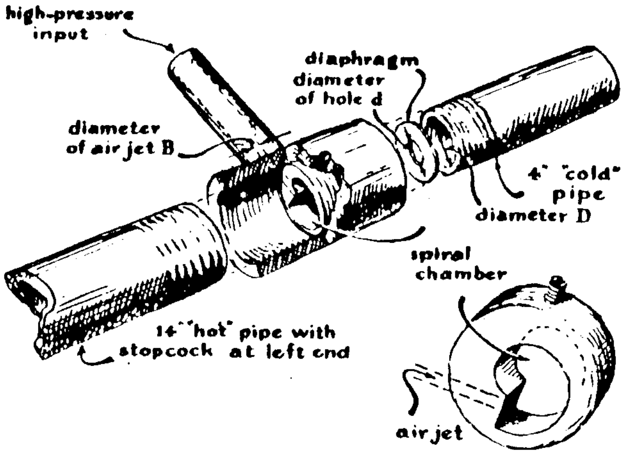
publ. Scientific American ? p. 515-520

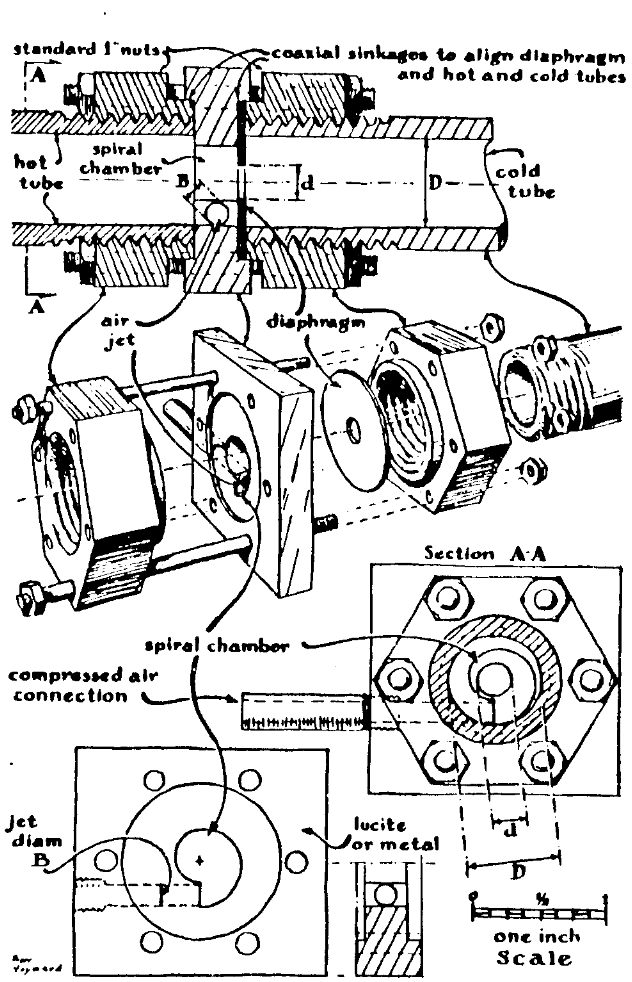
be devised if we could capture a small demon and train him to open and close a tiny valve. The demon would open the valve only when a fast molecule approached it, and close the valve against slow molecules. The water coming out of the valve would thus be hot. To produce a stream of cold water the demon would open the valve only for slow molecules.

Maxwell's demon would circumvent the law of thermodynamics which says in essence: "You can't get something for nothing." That is to say, one cannot separate cold water from hot without doing work. Thus when physicists heard that the Germans had developed a device which could achieve low temperatures by utilizing Maxwell's demon, they were intrigued, though obviously skeptical. One physicist investigated the matter at first hand for the U. S. Navy. He discovered that the device was most ingenious, though not quite as miraculous as had been rumored.

It consists of a T-shaped assembly of pipe joined by a novel fitting, as depicted in Figure 234. When compressed air is admitted to the "leg" of the T, hot air comes out of one arm of


high-pressure input
diaphragm diameter of hole d
diameter of air jet B
4" "cold" pipe diameter D
spiral chamber
14" "hot" pipe with stopcock at left end
air jet

234
Cutaway view of Hilsch tube showing relation of spiral chamber to central pipe-coupling. Detail of spiral at lower right



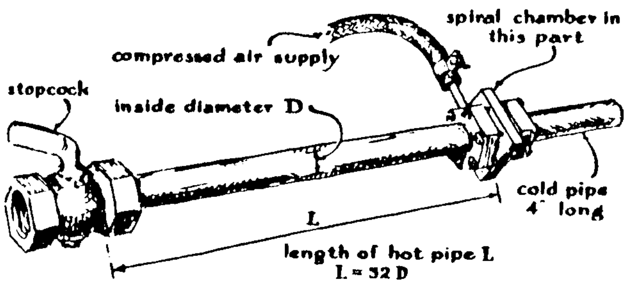
235
Version of Hilsch tube designed by Roger Hayward of Pasadena, Calif., for construction by amateurs. Note scale, lower right

the T and cold air out of the other arm! Obviously, however, work must be done to compress the air.

The origin of the device is obscure. The principle is said to have been discovered by a Frenchman who left some early experimental models in the path of the German Army when France was occupied. These were turned over to a German physicist named Rudolf Hilsch, who was working on low-temperature refrigerating devices for the German war effort. Hilsch made some improvements on the Frenchman's design, but found that it was no more efficient than conventional methods of refrigeration in achieving fairly low temperatures. Subsequently the device became known as the Hilsch tube.

The Hilsch tube may be constructed from a pair of modified nuts and associated parts as shown in Figure 235. The horizontal arm of the T-shaped fitting contains a specially machined piece, the outside of which fits inside the arm. The inside of the piece, however, has a cross section which is spiral with respect to the outside. In the "step" of the spiral is a small opening which is connected to the leg of the T.

Thus air admitted to the leg comes out of the opening and spins around the one-turn spiral. The "hot" pipe is about 14 inches long and has an inside diameter of half an inch. The far end of this pipe is fitted with a stopcock which can be used to control the pressure in the system [*see Fig. 236*]. The "cold" pipe is about four inches long and also has an inside diameter of half an inch. The end of the pipe which butts up against the spiral piece is fitted with a washer, the central hole of which is about a quarter of an inch in diameter. Washers with larger or smaller holes can also be inserted to adjust the system.



236
Relative lengths of hot and cold tubes of Hilsch device

Three factors determine the performance of the Hilsch tube: the setting of the stopcock, the pressure at which air is admitted to the nozzle, and the size of the hole in the washer. For each value of air pressure and washer opening there is a setting of the stopcock which results in a maximum difference in the temperature of the hot and cold pipes [*see Fig. 237*]. When the device is properly adjusted, the hot pipe will deliver air at about 100 degrees Fahrenheit and the cold pipe air at about −70 degrees (a temperature substantially below the freezing point of mercury and approaching that of "dry ice"). When the tube is adjusted for maximum temperature on the hot side, air is delivered at about 350 degrees F. It must be mentioned, however, that few amateurs have succeeded in achieving these performance extremes. Most report minimums on the order of −10 degrees and maximums of about +140 on the first try.

Despite its impressive performance, the efficiency of the Hilsch tube leaves much to be desired. Indeed, there is still disagreement as to how it works. According to one explanation, the compressed air shoots around the spiral and forms a high-velocity vortex of air. Molecules of air at the outside of the vortex are slowed by friction with the wall of the spiral. Because these slow-moving molecules are subject to the rules of centrifugal force, they tend to fall toward the center of the vortex.

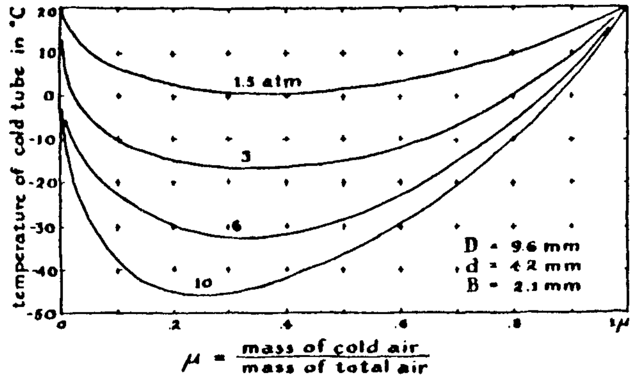
The fast-moving molecules just inside the outer layer of the vortex transfer some of their energy to this layer by bombarding some of its slow-moving molecules and speeding them up. The net result of this process is the accumulation of *slow-moving, low-energy* molecules in the center of the whirling mass, and of *high-energy, fast-moving* molecules around the outside. In the thermodynamics of gases the terms "high energy" and "high velocity" mean "high temperature." So the vortex consists of a core of cold air surrounded by a rim of hot air.

The difference between the temperature of the core and that of the rim is increased by a secondary effect which takes advantage of the fact that the temperature of a given quantity of gas at a given level of thermal energy is higher when the gas is confined in a small space than in a large one; accordingly when gas is allowed to expand, its temperature drops. In the case of the Hilsch tube the action of centrifugal force compresses the hot rim of gas into a compact mass which can escape only by flowing along the inner wall of the "hot" pipe in a compressed state, because its flow into the cold tube is blocked by the rim of the washer.

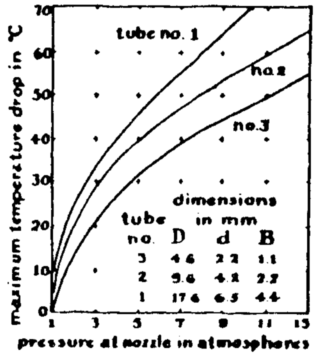
The amount of the compression is determined by the adjustment of the stopcock at the end of the hot pipe. In contrast, the relatively cold inner core of the vortex, which is also considerably above atmospheric pressure, flows through the hole in the washer and drops to still lower temperature as it expands to atmospheric pressure obtaining inside the cold pipe.

Apparently the inefficiency of the Hilsch tube as a refrigerating device has barred its commercial application. Nonetheless amateurs who would like to have a means of attaining relatively low temperatures, and who do not have access to a supply of dry ice, may find the tube useful. When properly made it will deliver a blast of air 20 times colder than air which has been chilled by permitting it simply to expand through a Venturi tube from a high-pressure source. Thus the Hilsch tube could be used to quick-freeze tissues for miscroscopy, or to chill photomultiplier tubes. But quite apart from the tube's potential application, what could be more fun than to trap Maxwell's demon and make him explain in detail how he manages to blow hot and cold at the same time?

Incidentally, this is not a project for the person who goes in for commercially made apparatus. So far as I can discover Hilsch tubes are not to be found on the market. You must make your own. Nor is it a project for the experimenter who makes a specialty of building apparatus from detailed specifications and drawings. The dimensions shown in the accompanying figures are only approximate. Certainly they are not optimum values. But if you enjoy exploration, the device poses many questions. What would be the effect, for example, of substituting a divergent nozzle for the straight one used by Hilsch? Why not create the vortex by impeller vanes, such as those employed in the stator of turbines? Would a spiral chamber in the shape of a torus improve the efficiency? What ratio should the diameter of the pipes bear to the vortex chamber and to each other? Why not make the spiral of plastic, or even plastic wood? One can also imagine a spiral bent of a strip of brass and soldered into a conventional pipe coupling. Doubtless other and far more clever alternatives will occur to the dyed-in-the-wool tinkerer.



238
Performance graph of Hilsch tube showing effect of hot-tube stopcock adjustment for various input pressures



237
Graph of performance for three tubes made by Rudolf Hilsch. The tabulated dimensions "D" "d" and "B" refer to Figure 238

VORTEC CORPORATION

VORTEX TUBES

Refrigeration from Compressed Air

What Is a Vortex Tube?

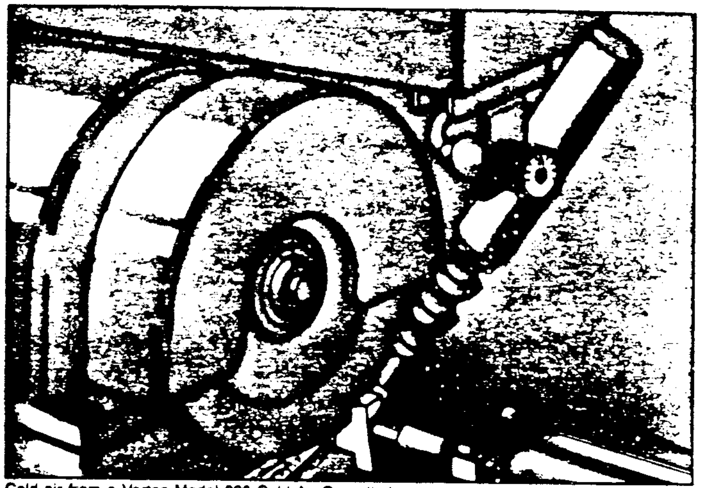
A Vortex tube is an instrument with no moving parts that converts a supply of ordinary compressed air into two airstreams at slightly above atmospheric pressure, one hot and one cold. A Vortex tube can produce up to 6000 BTUH of refrigeration or temperatures down to −40°F, using nothing more than compressed air at 100 psig. A valve in the hot air outlet allows you to infinitely adjust the flows and temperatures over a wide range.

Benefits of Vortex Tubes

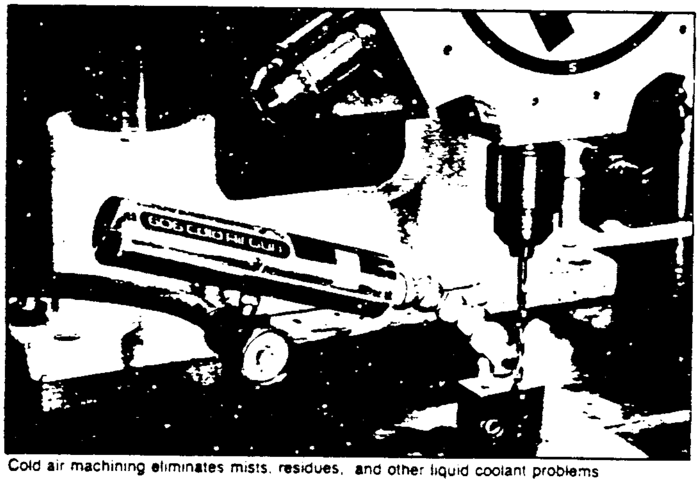
- No moving parts
- Virtually maintenance free
- Low initial cost
- Small size
- Highly reliable
- Easily controlled
- Lightweight
- Portable
- No spark/explosion hazard
- Spot cools without waste
- Instant on/off
- No RF interference

How Is Industry Using Vortec Tubes?

- Cool machining operations
- Set solders, adhesives
- Temperature cycle thermostats, instruments and electronic components
- Dehumidify gas samples
- Cool electronic control cabinets
- Cool workers in protective helmets, hoods and suits
- Test automobile temperature sensors and chokes
- Cool industrial sewing needles
- Chill environmental chambers
- Cool mold tooling
- And many more...see the application cases in section I.



Cold air from a Vortec Model 606 Cold Air Gun eliminates cracks during tool grinding.



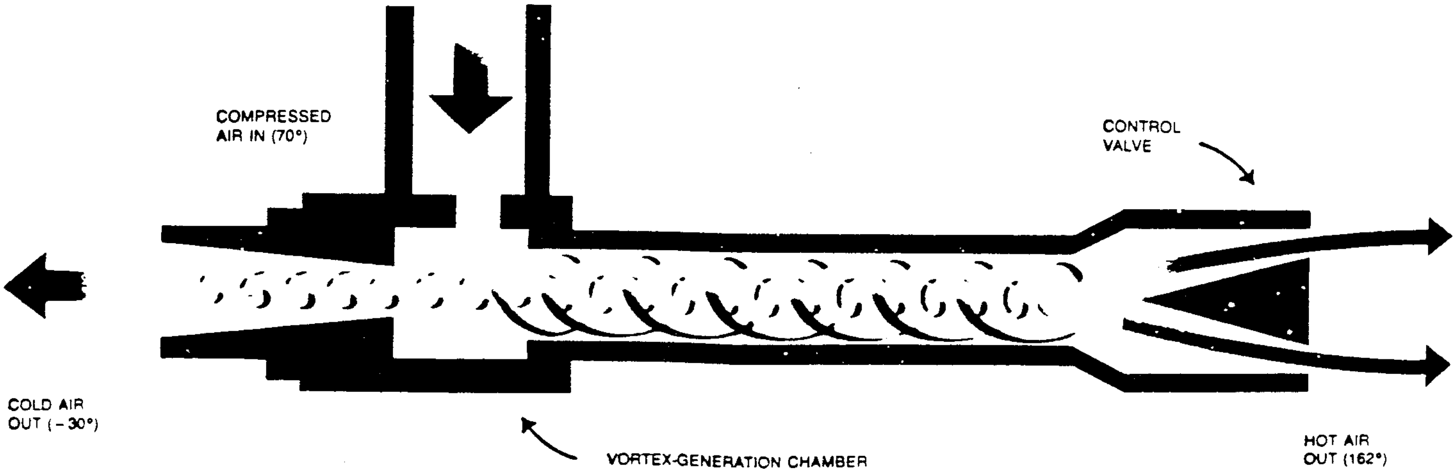
Cold air machining eliminates mists, residues, and other liquid coolant problems

How Does a Vortex Tube Work?

High pressure air enters a stationary generator and is released with a Vortex motion, spinning along the tube's walls toward the hot air end at sonic speeds up to 1,000,000 rpm. Air near the surface of this spinning vortex becomes hot, and some of it exits through the needle valve in the hot end. The air that does not escape through the hot air needle valve is forced back through the center of the warm air vortex.

Because the air forced back through the sonic-velocity hot airstream moves at a slower speed, a simple heat exchange takes place. The inner, slower-moving column of air gives up heat to the outer, faster-moving column. When the slower, inner column of air exits through the center of the stationary generator and out the cold exhaust, it has attained an extremely low temperature.

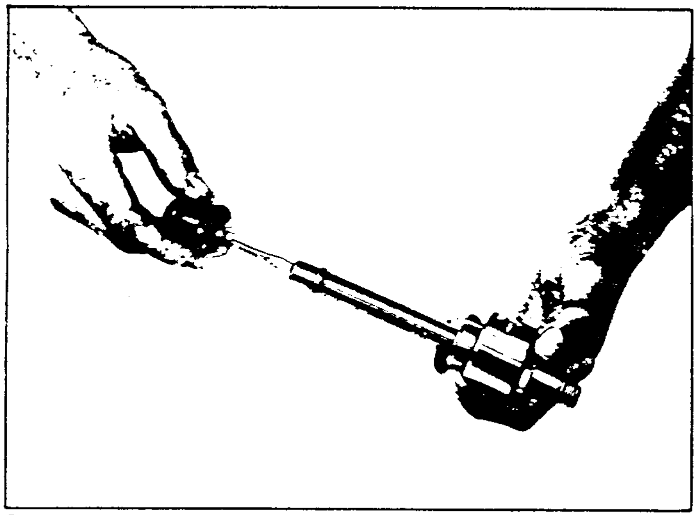
For a more detailed explanation of this process, initially discovered in 1930, read the "Short Course" in section H or call one of our Application Engineers on the Hot Line.



Vortex Tube Performance

You can vary the volume and temperature of the cold air produced by a Vortex tube by adjusting the valve in the hot air outlet. This controls what is known as the "cold fraction," which is the percent of the input air volume released through the cold outlet. For example, if the total volume of compressed air input is 15 SCFM and the cold fraction is 70%, then 10.5 SCFM exits the cold end and 4.5 SCFM exits the hot end. A *high cold fraction* (more than 50% of the input air released through the cold air end) produces *maximum refrigeration*—the greatest BTU output. This usually occurs at cold fractions of 60%-70%, where the mass of air released at the cold outlet and the temperature drop of the air are optimized to yield maximum efficiency and BTU's. Most applications, such as cooling electrical controls, liquid baths, and personnel, use high cold fractions for maximum refrigeration.

A *low cold fraction* (less than 50% of input air released through the cold end) produces the *lowest temperatures,* but with a reduced flow of air. A low cold fraction, with its extreme low temperature and low flow, is useful in cooling small machining operations, small parts, glass, testing of electronic components, and laboratory experiments.



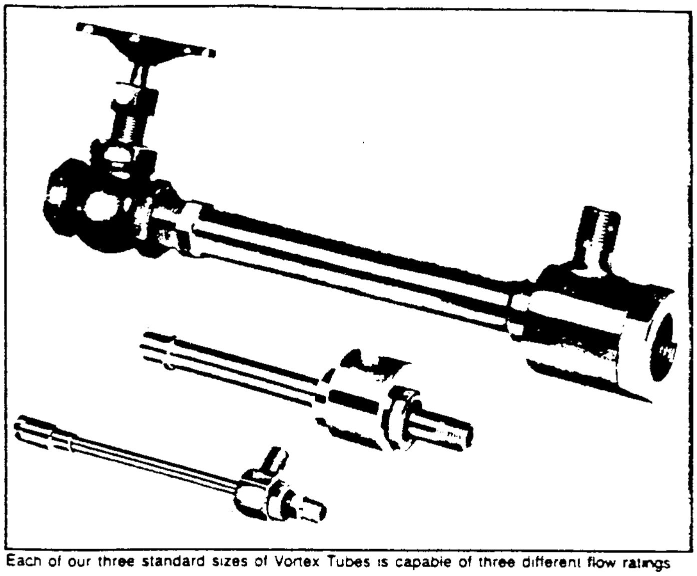
Temperature of cold air from a Vortex Tube is adjusted with a simple valve in the hot-air end.

Standard Models from Vortec

Vortec Corporation was the first company to fully develop and commercialize the Vortex tube.

In more than two decades of work, we have developed three standard tube sizes, each capable of three flow ratings. We accomplish this with three differently-sized, interchangeable generators which provide three CFM ratings for each tube.

The nine different capacities provided by our Vortex tubes cover the entire range of cooling capability which is economically practical with a Vortex tube. This enables you to size a unit for your application without wasteful over-capacity.

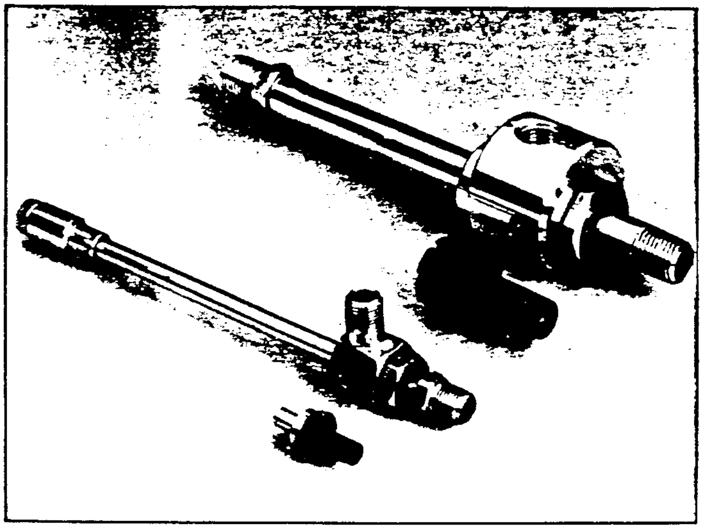


Each of our three standard sizes of Vortex Tubes is capable of three different flow ratings

Optimizing Vortex Tube Efficiency For Your Application

By changing the diameter of the cold air passage in the center of the vortex generator, you can configure your Vortex tube for maximum efficiency as a high-cold-fraction unit or a low-cold-fraction unit. We provide simple bushings to accomplish this in our 106 and 328 tubes. We designate these bushings H and L for high and low cold fractions. Each bushing is sized to match the CFM capacity of its corresponding vortex generator. For example, generators for a 106 tube (designated 2, 4, 8) may be used with bushings 2-H or 2-L, 4-H or 4-L, 8-H or 8-L, depending on the cold fraction you want.

The bushing and generator are separate parts for our 106 and 328 tubes. For the 208 tube, the bushing and generator are manufactured as a single part, with the model numbers again designating CFM rating and high or low fraction efficiency: 11-H, 11-L, 15-H, 15-L, 25-H, 25-L.

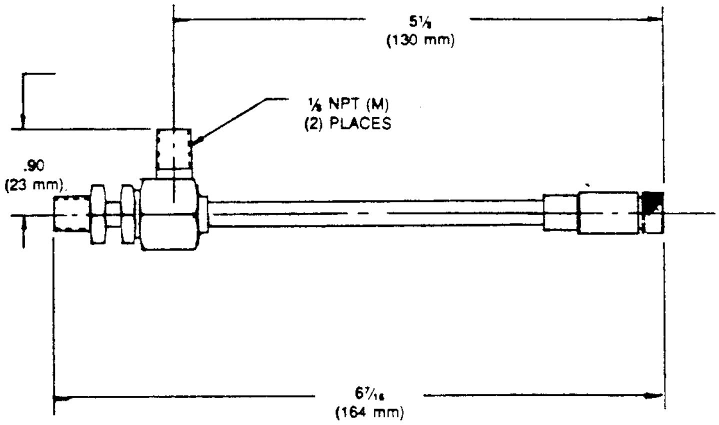


The stationary generators and bushings inside the Vortex Tube control air flow and cold-fraction capabilities.

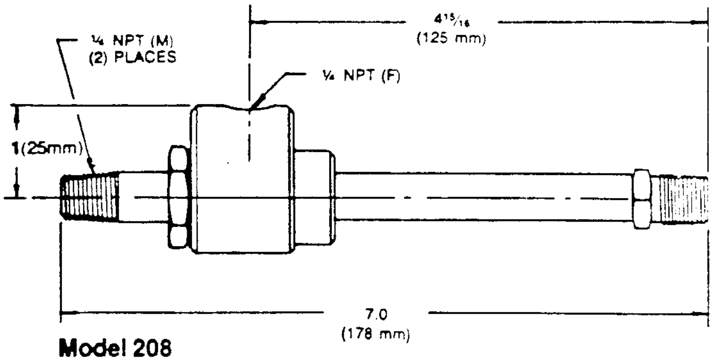
Vortex Tube Capacities

Model Number	Inlet Pressure (PSIG)*	Air Consumption (SCFM)	Capacity (BTU/H)
106-2-H	100	2	100
106-4-H	100	4	200
106-8-H	100	8	400
208-11-H	100	11	600
208-15-H	100	15	900
208-25-H	100	25	1500
328-50-H	100	50	3000
328-75-H	100	75	4500
328-100-H	100	100	6000

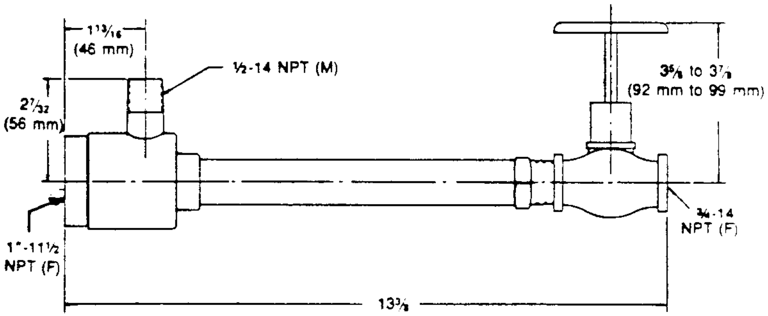
*Inlet temperature: 70°F.



Model 106



Model 208



Model 328

Specifying a Vortex Tube

This table shows the approximate temperature drops and rises you can expect from Vortex tubes adjusted to various cold fractions. The primary determinant in Vortex tube performance, in addition to the cold fraction setting, is inlet pressure.

The table assumes an inlet temperature of 70°F. Keep in mind that the cooling performance of the Vortex tube can be degraded by higher supply air temperatures, by moisture in the supply air which can condense and freeze (easily corrected with an air filter), or by excessive back pressure at either the hot or cold air outlet. A clean Vortex tube, operating with constant inlet pressure and temperature, will maintain temperatures within ±1°F.

Vortex Tube Performance Data

Inlet Pressure PSIG	Cold Fraction, %						
	20	30	40	50	60	70	80
20	61.5 14.5	59.5 24.5	55.5 36.0	50.5 49.5	[illegible] 64.0	[illegible] 82.5	27.5 107
40	[illegible] 20.5	86.0 35.0	80.0 51.5	73.0 71.0	[illegible] 91.5	[illegible] 117	[illegible] 147
60	104 23.5	100 40.0	[illegible] 58.5	[illegible] 80.0	[illegible] 104	[illegible] 132	[illegible] 168
80	115 25.0	110 43.0	102 63.0	92.0 86.0	[illegible] 113	[illegible] 143	[illegible] 181
100	123 26.0	118 45.0	110 66.5	[illegible] 91.0	[illegible] 119	[illegible] 151	[illegible] 192
120	[illegible] 26.0	124 46.0	[illegible] 69.0	[illegible] 94.0	[illegible] 123	[illegible] 156	[illegible] 195
140	[illegible] 25.5	[illegible] 46.0	121 70.5	[illegible] 96.0	[illegible] 124	[illegible] 156	[illegible] 193

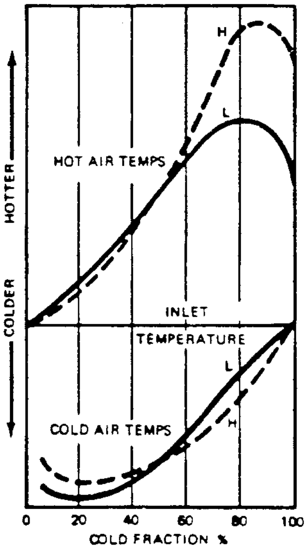
Figures in gray area give temperature drop of cold air, °F
Figures in white area give temperature rise of hot air, °F

Air Flow

The total discharge (hot and cold) in SCFM from a Vortex tube is proportional to absolute inlet pressure. To approximate the total flow for a given inlet pressure, use the following simplified formula:

$$\frac{(\text{inlet psig.} + 15) \times \text{cfm rating of generator}}{115} = \text{approx. total flow}$$

TEMPERATURE VERSUS COLD FRACTION FOR A TYPICAL VORTEX TUBE



Air Conditioning Power

To approximate the cooling and heating power in BTUH, use the following simplified formulas:

CF = Cold Fraction
CFM_t = Total Air Flow
CFM_c = Cold Air Flow = CFM_t (CF)
CFM_h = Hot Air Flow = CFM_t (100-CF)

t_i = Inlet Temperature
t_c = Cold Air Outlet Temperature
t_h = Hot Air Outlet Temperature

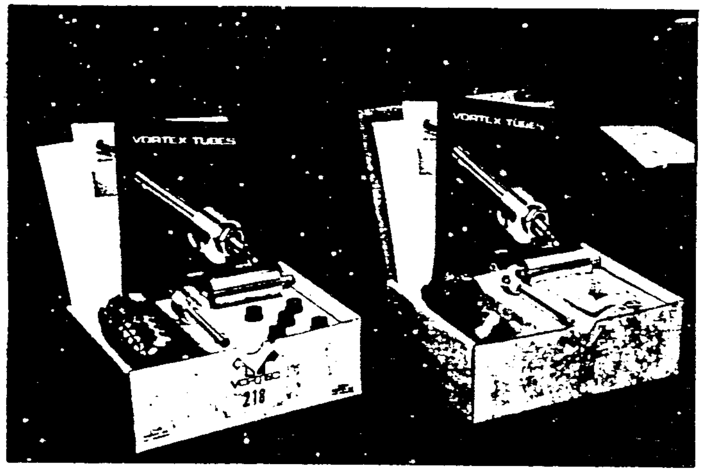
For Cooling:
$BTU/hr = 1.0746\,(CFM_c)\,(t_i - t_c)$
For Heating:
$BTU/hr = 1.0746\,(CFM_h)\,(t_h - t_i)$

Getting Started

The technical and practical questions you may have about designing a Vortex tube into a product or process can be handled by our Vortecnology Application Engineers. They not only know our products inside-out, but they've also helped thousands of customers use those products in a variety of industries and applications. Their experience is a technical resource that's available to you for the asking. And it's *FREE*.

Experimental Kits

If you want to "learn on your own," or if you need to periodically test Vortex tubes in your products or systems, we have available two experimental kits containing the generators and bushings needed to produce the full range of flows and temperatures with the two most commonly used tubes: the 106 and 208. The kits also contain a 5-micron filter, a cold-end muffler, and a booklet on the operation and application of Vortex tubes. The kits are handy design tools to have whenever you are contemplating the use of a Vortex tube in your product or system.



Experimental kits contain all parts necessary to produce the full range of flows and temperatures from the 106 and 208 Vortex Tubes

Specifications

	116 EXPERIMENTAL OUTFIT	218 EXPERIMENTAL OUTFIT
Vortex Tube	106	208
Refrigeration	up to 400 BTU/hr.	up to 1500 BTU/hr.
Flow Rates (SCFM)	2, 4, 8	11, 15, 25
Temperature	to −40°F	to −40°F

Accessories

We stock a full line of accessories that have proven the most-useful and most-asked-for by Vortex tube purchasers and system designers:

- Bushing/generator kits
- Cold end mufflers
- Hot end mufflers
- Thermostats
- Solenoid valves
- Air lines
- Pressure regulators
- Filter/separators
- and more

PRICE #106 $188—
#208 $339—

APPLIED VORTEX TUBE PRODUCTS

Standard System Packages

We've learned from working with our customers that a number of Vortex tube applications are almost universal, so we've packaged several systems with the tubes and accessories designed for specific uses. Four of those systems are described in this section: Cold Air Gun, Circuit Tester, Choke Tester, and Thread Guard. Two others—Vortex Control Cooling Systems and Mancooling—have grown into complete product lines. They are described fully in the sections devoted to those applications.

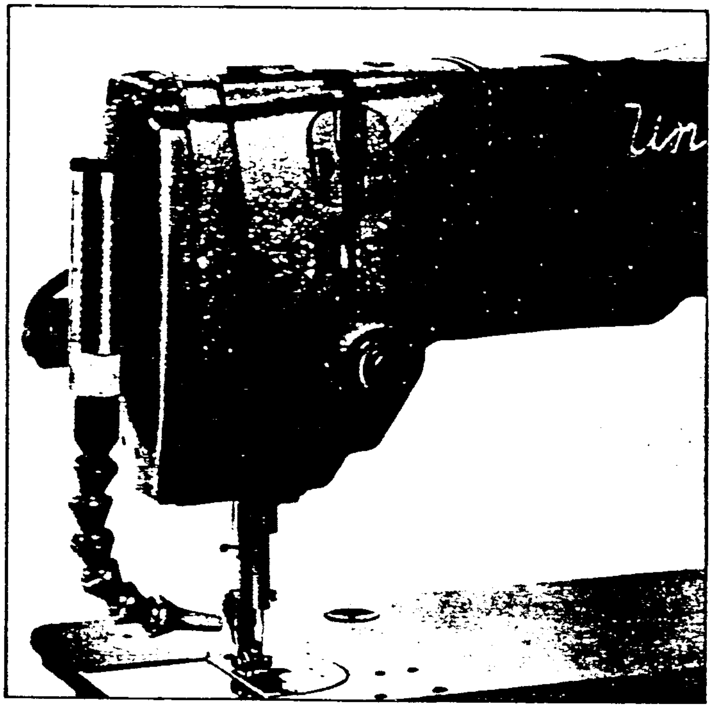
Thread Guard

Add the model 424 Thread Guard to any sewing machine and watch production soar, even in continuous operations for waistbands and belt loops, and with tough materials such as denim and belting.

The continuous stream of 10°F air from the Thread Guard virtually eliminates thread burning and breakage, as well as heat-related needle breakage. It's especially effective with synthetic threads.

Ready to Use

- Uses just 4 SCFM of shop air at 80-100 psig.
- Easily and quickly mounted.
- Adapts to any machine.
- Includes mounting hardware, five-micron filter, shut-off valve, pressure regulator/gauge, cold air tube and complete instructions.



Thread Guard cools sewing operations. It stops thread burning and breakage as well as heat-related needle breakage

VORTEC
1-800-441-7475

A Short Course on Vortex Tubes and Application Notes

Contents

History
Air Movement in a Vortex Tube
Vortex Tube Performance
Temperature Separation Effects in a Vortex Tube
Effects of Inlet Temperature
Using the Performance Table
The Heat Balance Formula
Air Flow at Various Inlet Pressures
Humidity
The Air Supply
Preparing the Air
Settings
Using the Cold Air
Noise Muffling
Maintenance

History

The vortex tube effects were first observed by Georges Ranque, a French physicist about 1930. He formed a small company to exploit the item but it failed soon. He presented a paper on the vortex tube to a scientific society in France in 1933, but it was met with disbelief and disinterest. Thereafter the vortex tube disappeared for several years, until Rudolph Hilsch studied it and published his findings in the mid-1940s.

Hilsch's paper stirred much interest where Ranque's had not. So much so, in fact, that most readers thought Hilsch had invented the device, and it was popularly called the "Hilsch Tube."

Since then, the vortex tube has become much better known to technical people. There has been a slow but steady increase in research and publication on the subject around the world. Well over 100 serious studies have been published in the world's scientific and engineering journals, scattered so that it is hard to assemble more than a fraction of them. Many popular articles and commentaries have been published. Many engineering schools and industrial and scientific groups are working on the vortex tube.

Today vortex tubes serve in a wide variety of industrial applications, including cooling workers, cooling electrical and electronic equipment, and many process cooling applications.

Air Movement in a Vortex Tube

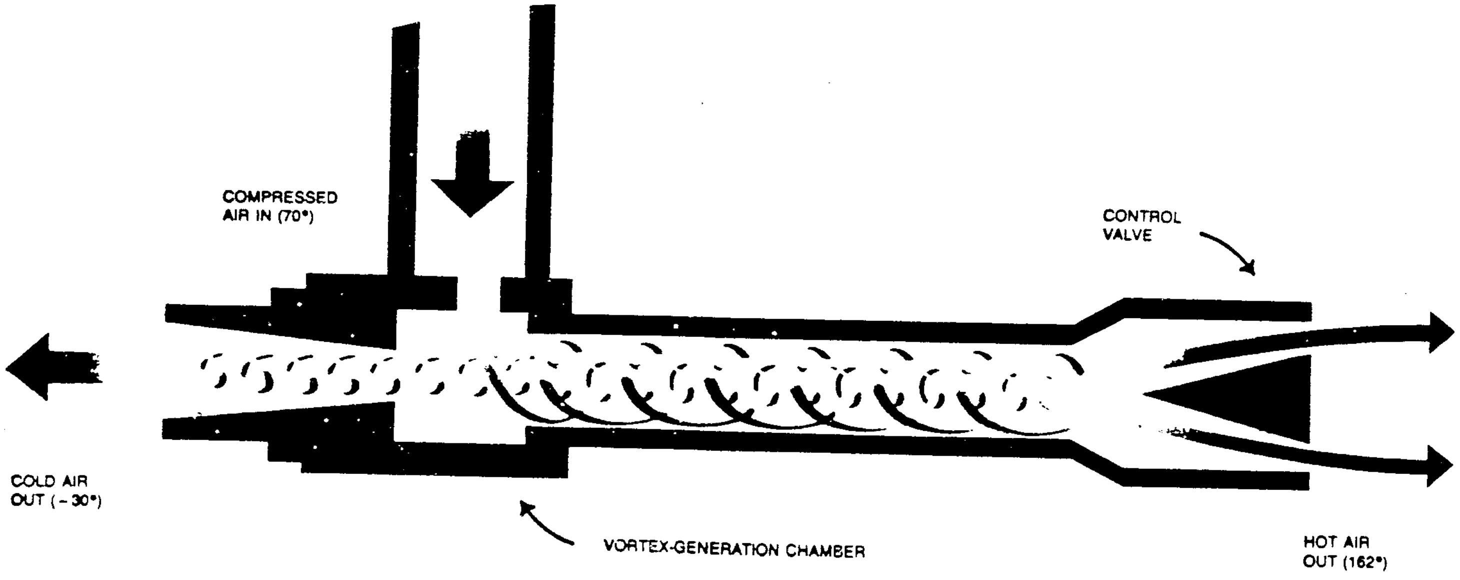
Below is a schematic drawing of a vortex tube showing the internal arrangement and the common names for certain important features.

High pressure air enters the inlet and enters the annular space around the generator. It then enters the nozzles where it loses part of its pressure as it expands and gains sonic or near-sonic velocity. The nozzles are aimed so that the air is injected tangentially at the circumference of the vortex generation chamber. All of the air leaves the vortex generation chamber and goes into the hot tube. It makes this choice (between hot and cold ends) because the opening to the hot tube is always larger than the opening to the cold tube (through the center of the generator). Centrifugal force keeps the air near the wall of the hot tube as it moves toward the valve at the end.

By the time the air reaches the valve it has a pressure somewhat less than the exit pressure at the nozzles, but more than atmospheric (assuming cold outlet is at atmospheric pressure). *It is always true that the pressure just behind the control valve is higher than the cold outlet pressure.*

The position of the valve determines how much air leaves at the hot end. For hot-cold separation, it must allow only part of the air to escape. The remaining air is forced to the center of the hot tube where, still spinning, it moves back toward the cold outlet. It goes all the way through the hot tube, through the center of the vortex generation chamber, to the cold outlet.

Remember the original stream of air in the hot tube did not occupy the center of the tube because of certrifugal force. Therefore it defines an ideal path for the inner stream to follow. This, combined with the above mentioned pressure difference between the valve and the cold outlet, is the reason there are two distinct spinning streams, one inside the other moving in opposite linear directions in the hot tube.



Vortex Tube Performance

As the valve position is changed, the proportions of hot and cold air change, but the total flow remains the same. Thus the amount of air exiting the cold end can be varied over a wide range for a given vortex tube. The amount of this air is known as the "cold fraction."

As you can imagine, one of the secrets of good vortex tube design is to avoid mixing of any of the cold inner stream (the cold fraction) with the warm or hot outer stream. If a tube is operating at a high cold fraction, the passage in the center of the generator must be large enough to handle the cold flow. If not, it will cause some of the cold air to be deflected away and mixed in with the warm air stream, thus wasting refrigeration. At low cold fractions the desired result is usually a small stream of very cold air. An opening too large will invite entrainment of some of the nearby warm air and raise the cold outlet temperature.

Thus for any given vortex tube of a fixed total flow capacity *there is an ideal opening size for every cold fraction.* Practically, a vortex tube user will normally want one of two *modes* of operation. Either *maximum refrigeration* (which occurs at about 70% cold fraction) or *lowest possible cold temperature* (which occurs at about 20% cold fraction). Accordingly Vortec offers H (high cold fraction) bushings designed with the optimum opening for maximum refrigeration and L (low cold fraction) bushings with the optimum smaller opening to create lowest possible cold temperatures.

Each of Vortec's standard tubes can also be fitted with generators for different CFM capacities. Thus we must offer an *H and L bushing for each CFM capacity* in a given tube. So the bushings must be selected *based on two parameters, capacity and mode.* This is why we adopt the simple nomenclature 2-H, 4-L, 8-H, etc.

The 328 tube is made with separate generators and bushings, but it is so rare that anyone would want a tube as large as this to create extremely low temperatures, that we don't show L versions in the price list. All listed 328 series tubes have H style bushings, but L style can be provided on a special order basis.

Temperature Separation Effects In a Vortex Tube

We have already covered the movement of air in the vortex tube. Now we shall attempt to explain why the hot air gets hot and the cold air gets cold.

You'll recall that the air in the hot tube has a complex movement. An outer ring of air is moving toward the hot end and an inner core of air is moving toward the cold end. Both streams of air are rotating in the same direction. More importantly, both streams of air are rotating at the *same angular velocity.* This is because intense turbulence at the boundary between the two streams and throughout both streams locks them into a single mass so far as rotational movement is concerned.

Now the proper term for the inner stream would be a *"forced vortex."* This is distinguished from a *"free vortex"* in that its rotational movement is controlled by some outside influence other than the conservation of angular momentum. In this case the outer hot stream forces the inner (cold) stream to rotate at a constant angular velocity.

In the bathtub whirlpool situation (which most people associate with the word "vortex"), a free vortex is formed. As the water moves inward, its rotational speed increases to conserve angular momentum. Linear velocity of any particle in the vortex is inversely proportional to its radius. Thus, in moving from a radius of one unit to a drain at a radius of ½ unit, a particle doubles its linear (tangential) speed in a free vortex. In a forced vortex with constant angular velocity, the linear speed decreases by half as a particle moves from a radius of 1 unit to a drain at a radius of ½ unit.

So, for the situation above, particles enter the drain with 4 times the linear velocity in a free vortex compared with a forced vortex. Kinetic energy is proportional to the square of linear velocity, so the particles leaving the drain of the forced vortex have 1/16th the kinetic energy of those leaving the drain of the free vortex in this example.

Where does this energy (15/16 of the total available kinetic energy) go? Therein lies the secret of the vortex tube. The energy leaves the inner core as heat and is transmitted to the outer core.

Now you might say the air in the cooling inner stream had to travel through the outer (heating) stream first. Why doesn't it heat the same amount it cools with no net cooling effect? Keep in mind that the rate of flow in the outer stre is always larger than that of the inner stream, since part of the outer stream is being discharged at the hot valve. If the BTUs leaving the inner stream equal the BTUs gained by the outer stream, the temperature drop of the inner stream must be more than the temperature gain of the outer stream because its mass rate of flow is smaller.

If this precept is clear in your mind, a little reflection will allow you to understand why hot end temperatures increase as cold fraction increases, and cold end temperatures decrease as cold fractions decrease.

Effects of Inlet Temperature

It is very easy to predict the temperature drops and rises in a vortex tube for various inlet temperatures. The basic rule to remember is that temperature drops or gains are *proportional to absolute inlet temperature.* Any temperature expressed in degrees Fahrenheit can be converted to absolute (degrees Rankine) by adding 460. That is, 0°F = 460°R or 70°F = 530°R.

Thus, the entire table is based on an inlet temperature of 530°R. If absolute inlet temperature doubles, so does the temperature drop or gain. As an example, suppose you want to find the temperature drop associated with a vortex tube operating at 30% cold fraction and with a 100 psig, 200°F inlet.

1. Table gives 118° drop for 100 psig, 70°F inlet and 30% CF.
2. Ratio of absolute inlet temperature

$$\frac{200 + 460}{70 + 460} = \frac{660}{530} = 1.245$$

3. Drop given in table times ratio is 118 × 1.245 = 147°
4. Cold end temperature is 200° − 147° = 53°F

This ratio can be used just as well when the inlet temperature is lower than the 70°F on which the table is based. For example, if inlet temperature were 0°F, ratio would be

$$\frac{0 + 460}{70 + 460} = \frac{460}{530} = .87$$

In this case the temperature drop is reduced.

Exactly the same approach can be used to convert the temperature rises given in the table. They are greater for inlet temperatures higher than 70°F and smaller for inlets below 70°F.

One additional comment on this method should be made. It applies to the pressure range shown on the table only. Whenever pressures considerably higher than the table are involved, the Joule Thompson effect alters the results somewhat. This effect is small at pressures of 140 psig and below, and can be ignored as it is in the method given above. Joule Thompson cooling is the very slight cooling that takes place as gases are throttled.

Using the Performance Table

Two rather important limitations of the performance table in the catalog should be recognized.

First, the table would seem to imply that temperature drops and rises are related to inlet pressure. This is not quite true. They are related in a complex way to the *absolute pressure ratio* between inlet and cold outlet. The table is based upon the assumption that the *cold outlet is at atmospheric pressure.* For any other cold end pressures, the table cannot be used.

You can appreciate the variation in temperature drops and rises if you consider how quickly the absolute pressure ratio changes with changes in cold end pressure. A 90 psig inlet (105 psia) provides a 7 to 1 ratio when the tube exhausts to atmospheric pressure [illegible] psig or 15 psia). If inlet pressure remains the same and cold outlet pressure rises to only 15 psig, the ratio drops to 3.5 to 1.

Calculations of temperature rises and drops for pressures other than those shown on the table can be made, but they are beyond the scope of this Short Course. Refer any such problems to Vortec.

The Heat Balance Formula

A very handy formula results from the fact that the energy extracted from the cold air by the vortex tube appears in the hot air.

The formula is:

$$CF \times (t_i - t_c - JT) = (100 - CF) \times (t_h - t_i + JT)$$

where CF = cold fraction, %

t_i = inlet air temperature, °F

t_c = cold air temperature, °F

t_h = hot air temperature, °F

JT = Joule-Thompson temperature correction °F = 4°F at an inlet pressure of 100 psig

By using this formula, cold fraction can be computed from the readings of the three thermometers alone without having to measure any air flow. As an example, suppose $t_i = 100°F$, $t_c = 50°F$, $t_h = 300°F$. Substituting in the formula,

$$CF \times (100 - 50 - 4) = (100 - CF) \times (300 - 100 + 4)$$

Solving for CF, CF = 81.5%

Vortex tubes obey this formula very closely, regardless of their efficiency, provided only that the hot pipe be insulated.

The formula can be rearranged as follows:

$$CF = \frac{t_h - t_i + 4}{t_h - t_c} \times 100$$

This is its handiest form for computing cold fraction.

Humidity Effects

The vortex tube does not separate humidity between the hot and cold air. The absolute humidity of both cold and hot air, in grains/pound, is the same as that of the entering compressed air.

Moisture will condense and/or freeze in the cold air if its dew point is higher than its temperature. The following table shows the amount of moisture that air can hold in the saturated vapor state as a function of air temperature, at standard atmospheric pressure of 14.7 psia:

Temperature, °F	110	100	90	80	70	60	50	
Saturation*	375	295	217	154	111	77	54	
Temperature, °F	40	30	20	10	0	−10	−20	−30
Saturation*	37	24	15	9	5.5	3.2	1.8	1.0

*Saturation Moisture Content Grains/lb. Air

For example, the above table shows that if the moisture content is 14 gr./lb., condensation will begin when the temperature of the cold air falls below 19°F. At 5 gr./lb., condensation will begin at −1°F.

The saturation moisture content of compressed air at 100 psig is given in the following table:

Temperature °F	110	100	90	80	70	60	50	40	30	20
Saturation Moisture Content Grains/lb. Air	48	38	28	20	14	9.9	6.9	4.7	3.1	1.9

By comparing the two tables, it is possible to predict the amount of moisture in the compressed air, and the temperature at which moisture will begin to precipitate or freeze in the cold air. As an example, suppose the compressed air is aftercooled to 80°F following compression, and the precipitated water drained off. Then the second table shows that it will carry 20 grains/lb. of water vapor. When this expands in the vortex tube, the upper table shows that precipitation will begin in the cold air when its temperature falls below 26°F if its pressure is 14.7 psia.

If the compressed air is cooled under pressure by a chiller to 40°F, the second table shows that it will then carry 4.7 grains/lb of water vapor. When expanded in the vortex tube, precipitation will begin when the temperature of the cold air falls below −3°F at 14.7 psia.

If, under unusual conditions, some moisture precipitates in the cold air, the temperature of the cold air will thereby be caused to rise approximately 3/4°F for each grain of moisture that precipitates. This is because some of the sensible refrigeration of the cold air is consumed in producing latent refrigeration of the moisture. This refrigeration is not lost but reappears in the cold air as it warms up in performing its air conditioning duty after leaving the vortex tube, when the precipitated moisture re-evaporates.

The tables show that condensation will not normally occur at moderate cold end temperatures. When temperatures are low enough to cause condensation, it appears as snow. The snow has a sticky quality due to oil vapor and will gradually collect and block cold air passages. Continuous operation at low temperatures can be assured by means of an air dryer or injection of an antifreeze mist into the compressed air feeding a vortex tube.

When selecting dryers give consideration to refrigerative and deliquescent types. While their drying abilities are limited (and need to be considered) they are quite compatible with the vortex tube. Chemical dessicant dryers such as silica-gel and molecular sieve types are exothermic, and tend to heat the compressed air causing refrigeration losses.

Application Notes

The Air Supply

Pressure

Standard vortex tubes made by Vortec Corporation are designed to utilize a normal shop air supply of 80 to 110 psig pressure. Unless pressures run considerably higher than 110 psig, do not use a regulator to reduce the inlet pressure. Pressures higher than 250 psig must not be used. Pressures lower than 80 psig will still produce some cooling. However, both the temperature drops and the flows are reduced due to the lower inlet pressures.

Line Sizes

Up to 25 SCFM, runs of pipe less than 10 feet long may be 1/4" size without excessive pressure drop. Up to 50 feet use 3/8" pipe, and use 1/2" pipe over 50 feet. Rubber hose of suitable pressure rating may be used. Consider 3/8" id hose to be the same as 1/4" pipe, and 1/2" id hose to be the same as 3/8" pipe. Remember that lower transmission pressures will exhibit even greater pressure drops, so care must be taken to avoid large losses in the inlet air piping.

Compressor Size

In most large plants, the size of the compressor is adequate to handle many vortex tubes operating simultaneously. For smaller plants, estimate horsepower required based upon the rated capacity of the tubes. For a 100 psig system, it takes one horsepower to compress four SCFM of air.

Preparing the Air

Moisture

All compressed air systems will have condensed water in the lines unless a dryer is in use. To remove condensed water from the air, a filter-separator *must be used.* Automatic drain types are recommended unless the area is always tended by a responsible employee who can empty the collection bowl periodically. Place the filter-separator as near to the vortex tube as possible.

Dryers

Normally a dryer is not required for vortex tube applications. Occasionally, however, when very low outlet temperatures are produced, icing will cause problems. Also, some applications may require the cold air stream to be completely free of condensed water or ice. A chemical dryer (silica gel, heatless, or other type) can be used in the inlet line to eliminate condensed water or ice in the cold air stream. The dryer should be rated to produce an atmospheric dew point lower than the lowest expected cold outlet temperature.

Dirt

Because of the water in compressed air lines, there is always rust and dirt present. Vortec's filter-separators effectively remove these contaminants by using a 5 micron filter. Replacement filters are available at nominal cost, and it is necessary for the user to determine the frequency of replacement based on the conditions prevailing in his plant.

Oil

Never use vortex tubes downstream of a lubricator. Oil in the air which has been introduced by the compressor lubrication system is usually not a problem for Vortec products, but occasionally older compressors produce very oily air. If the plant air is very oily, use an oil removal filter downstream of the filter-separator. The oil removal filter removes dirt, water, and oil aerosols with an effective filtration of 0.01 micron.

Settings

Maximum Refrigeration. Maximum refrigeration occurs when a vortex tube operates at 60 to 70 percent cold fraction. This is where the product of the mass of cold air and its temperature drop is the greatest. Many applications such as cooling electrical controls, liquid baths, and personal air conditioning use this maximum refrigeration setting. For maximum refrigeration, use H style bushings.

Minimum Temperature. Some applications require the lowest possible cold output temperature. Examples are cooling glass, cooling hot parts, and using cold air to cool machining operations. These air spraying applications usually work better with very cold air, and results seem not to depend upon the refrigeration rate. For these applications, L style bushings and cold fractions in the 20 to 40 percent range are best.

Using the Cold Air

Back Pressure. One of the most common mistakes with vortex tubes is to restrict the cold outlet. This will cause a loss of performance. A small back pressure on the cold outlet to allow the air to move through piping or ducting is acceptable, but back pressure, measured at the tube, should be limited to less than 5 psig. Keep in mind the tube is responsive to the absolute pressure ratio applied and back pressures as low as 15 psig cut this ratio in half. Some pressure is available at the hot end, and it can be used so long as compensating adjustments in the control valve settings are made.

Insulation. As with any thermodynamic device, the proper use of insulation will improve vortex tube system performance. Avoid ducting the cold air through large thermal masses such as heavy piping, drilled holes in large blocks, etc. If possible use plastic tubing or piping. Foam type insulation can also be quite helpful.

Noise Mufflers

General. A common misconception is that a vortex tube emits a scream or whistle due to the sonic speeds inside. Actually such noise is rarely observed, but the sound of escaping air is always present, and in some cases it must be muffled. Ordinarily the cold air will be ducted into an enclosure or through some pipe or tubing. This alone may reduce its noise level to acceptable limits. Hot air escapes in smaller amounts in most applications and may not be objectionable. Nevertheless, jets of escaping air can be quite objectionable if continued over a long period of time near worker. In such situations mufflers are available, and should be used.

Cold Muffling. Mufflers used on the cold air must not be of a stuffed or porous type. Their small openings will quickly block with ice which has condensed and frozen in the cold air stream. Baffle type mufflers and silencers are best for the cold air. Avoid selecting any muffler which will apply high back pressures to the vortex tube.

Hot Muffling. Nearly any air silencer or muffler will work on the hot end. One should avoid selecting a muffler made from plastics or other materials with low resistance to heat since hot end temperatures can easily exceed 200°F.

Maintenance

Since vortex tubes have no moving parts, they are highly reliable, and require little or no maintenance. Prolonged use with dirty or oily air can cause wear or dirt collection in the tube. Occasional disassembly, inspection, and cleaning are the only maintenance activities required.

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Journal of the ASRE

An Analysis of the Hilsch Vortex Tube

D. S. Webster

Engineering Research Laboratory
E. I. du Pont de Nemours & Co., Inc.

The American Society of Refrigerating Engineers takes pleasure in presenting this analysis of the Hilsch vortex tube together with pertinent discussions of the subject matter. Although the analysis is controversial, it is felt that Mr. Webster's theory, together with the ensuing discussions, form a material contribution toward a better understanding of this interesting refrigerating device. There are already indications that the Hilsch tube may emerge from the category of a laboratory toy to become a practical device with reasonably economical operation in some specialized applications.

THE May 1946 issue of *Industrial and Engineer Chemistry*[4] carried a short account of a small, cyclone-type air jet, or "vortex tube," used to obtain low temperatures. Originally suggested by M. G. Ranque,[10] it had been in operation in the laboratory of Dr. Hilsch, in Germany, and was discovered in July 1945 by an American investigation group. Great interest was stirred up by the account because of the seemingly paradoxical behavior of the vortex tube. Several types of hypothesis[1,2,6] were advanced to explain how hot air could come out of one end and cold air out the other end of the apparently simple tube. The Du Pont Engineering Research Laboratory became interested in the device, carried out a limited investigation of its refrigerating properties, and developed a hypothesis to explain these properties. The hypothesis and pertinent data are presented in this paper. A recently published article on the vortex tube by G. Burkhardt[9] only slightly overlaps the subject matter of this paper.

It should be pointed out and emphasized that this work was not "research" in the generally accepted meaning of the term; it was, rather, a high-spot investigation to make certain that no previously unrecognized principles of an important nature were being overlooked. As a consequence of this situation, many of the elements of a "research study" were not included; for example, a detailed experimental study should be made of the flow patterns inside the tube. Nevertheless, it is felt that the speculations presented here may be of some value to investigators interested in developing the device further, even if the only value is to stimulate them into proving the speculations wrong.

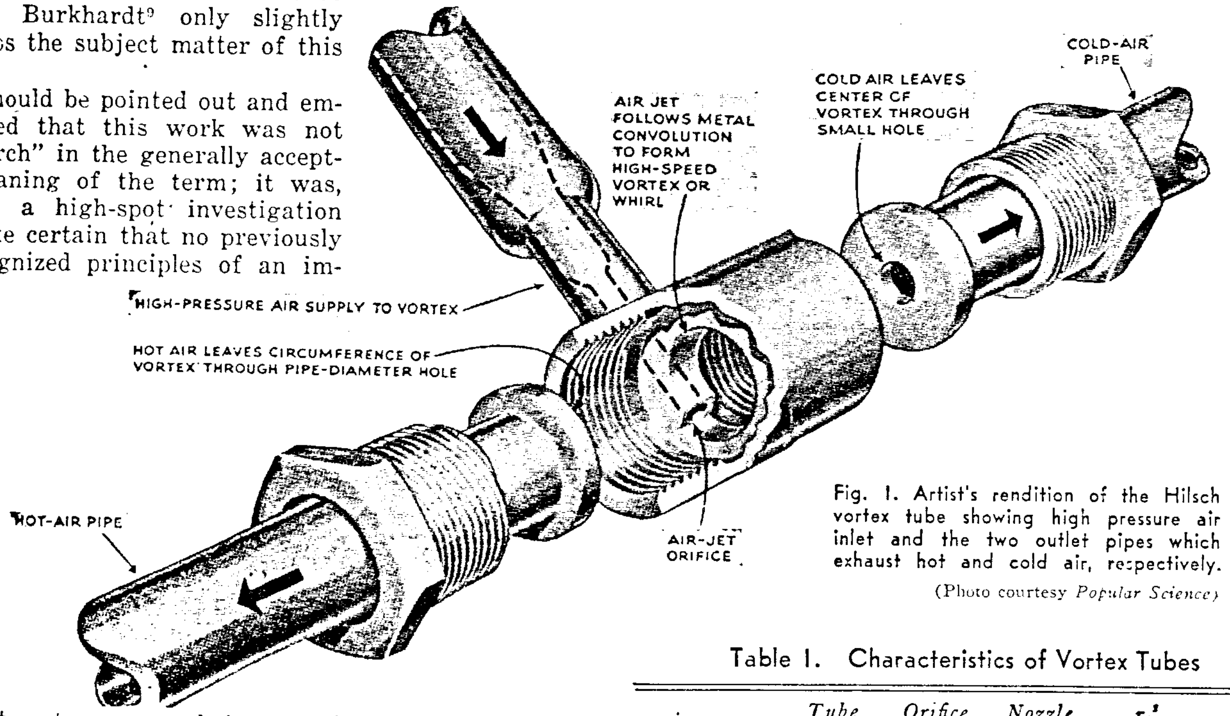

AIR JET FOLLOWS METAL CONVOLUTION TO FORM HIGH-SPEED VORTEX OR WHIRL
COLD AIR LEAVES CENTER OF VORTEX THROUGH SMALL HOLE
COLD-AIR PIPE
HIGH-PRESSURE AIR SUPPLY TO VORTEX
HOT AIR LEAVES CIRCUMFERENCE OF VORTEX THROUGH PIPE-DIAMETER HOLE
HOT-AIR PIPE
AIR-JET ORIFICE

Fig. 1. Artist's rendition of the Hilsch vortex tube showing high pressure air inlet and the two outlet pipes which exhaust hot and cold air, respectively.

(Photo courtesy *Popular Science*)

Description of Hilsch Vortex Tube

Figure 1 is an illustration of the Hilsch Vortex Tube taken from the May 1947 issue of *Popular Science*. The critical dimensions of several models used by Hilsch are given in Table 1. If further details on construction are desired, see References 2 and 4.

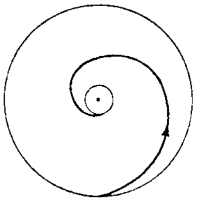
Compressed air enters the tube tangentially through a simple nozzle. The point of entry is beneath the step of a single spiral turn, used to help obtain rotational symmetry of the gas in the tube. The center of rotation is the same as that of the "circle" described roughly by the upper portion of the spiral, so that the incoming air enters under the rotating mass and joins it gradually,

Table 1. Characteristics of Vortex Tubes

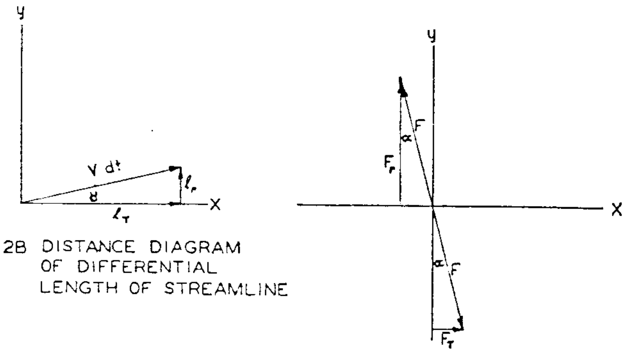
Designation	*Tube ID,* D_w	*Orifice diam,* D_o	*Nozzle diam,* D_n	$Z = \frac{r_o^2}{r_w^2}$	*Comment*
E.R.L.-Lucite	8.7 mm	2.4 mm	1.6 mm	0.074	No spiral
Hilsch-1-c	4.6 mm	2.2 mm	1.1 mm	0.23	Std (Fig 1)
Hilsch-2	9.6 mm	4.3 mm	2.3 mm	0.20	Std (Fig 1)
Hilsch-3	17.6 mm	6.5 mm	4.1 mm	0.136	Std (Fig 1)

This paper presented at the 45th Annual Meeting of the AMERICAN SOCIETY OF REFRIGERATING ENGINEERS, Dec. 4 to 7, in Chicago.

rather than abruptly. The hot-air pipe and the cold-air pipe butt against the sides of the spiral, located in such a way that the rotational axis of the gas is in line with the geometric axes of the pipes; in addition, the radii of the pipes are the same as that of the circle referred to above, so that there is essentially no change in rotation as the gas enters the hot pipe. The end of the cold pipe in contact with the spiral is closed with a diaphragm that has an orifice in its center, so that gas leaving through the cold pipe is taken from the center of the rotating mass. The end of the hot pipe adjacent to the spiral has no such diaphragm. Ordinarily, the vortex motion would suck air through the orifice into its center, but this is prevented by closing off the throttling valve on the hot pipe until air begins to pass *out* through the orifice.



2A
SINGLE SPIRAL
STREAMLINE IN ROTATING MASS



2B DISTANCE DIAGRAM OF DIFFERENTIAL LENGTH OF STREAMLINE

2C FORCE DIAGRAM FOR DIFFERENTIAL LENGTH OF STREAMLINE

Fig. 2. Diagrams indicating the analysis in this paper of the Hilsch tube.

A simple modification of the Hilsch vortex tube was made at the Engineering Research Laboratory by drilling a 11/32-in. hole through opposite faces of a 1½-in. tube of Lucite acrylin resin. A 1/16-in. hole was then drilled so that it entered the original hole tangentially, at a point equidistant from the two faces. Since this was the jet nozzle, the outer end was enlarged and threaded and connected to a compressed air line. Each end of the 11/32-in. hole was also enlarged and threaded for a distance of ½-in. A brass disk with a 3/32-in. orifice drilled through its center was placed in one end, and a pipe of approximately 11/32-in. ID, 6 in. long, was screwed in until it pushed the disk against the internal shoulder. A similar pipe, 12 in. long, was screwed into the other side of the hole and fitted with a throttling valve at its far end. During operation, hot air came out the latter pipe, and cold air out the former (with orifice).

Refrigerating Properties of Vortex Tubes

The importance of what Hilsch later termed "rotational symmetry" became apparent to us when it was found that an improperly centered orifice could cause a considerable decrease in temperature fall. Also, the Lucite vortex tube, with inside bore inadvertently made slightly out of round, nearly doubled its temperature fall when drilled to a true cylinder. The importance of the single spiral turn in obtaining rotational symmetry is indirectly shown by the fact that our tube, made without it, caused only about two-thirds as much temperature fall as Hilsch's tubes (see Fig. 4).

Polishing the bore of the Lucite tube provided an additional 10 percent temperature fall, thus demonstrating that frictional drag of the rotating gas on the tube walls is still another factor influencing operation.

The length of the cold pipe was varied between 3 and 6 in., and the hot-pipe length between 8 and 15 in. without appreciably affecting performance. Apparently, the smaller figure in each of these sets was adequate. However, a hot pipe with twice the diameter of the rest of the Lucite vortex tube decreased the amount of cooling to about one-third, although a similar effect was not produced with the cold pipe.

Straightening vanes installed in either the hot or the cold pipe to diminish swirl, simply interfered with operation of the device to the point where there was practically no cooling.

Hilsch makes the additional observation that, for best performance, the nozzle and orifice should be as close together as possible.

Figures 3 and 4 present plots of data taken mainly from Hilsch[2] but including also some of our results. The tubes referred to have the dimensions given in Table 1.

As shown in Figure 4, performance of the Lucite tube approaches Hilsch's well-made tubes even though the Lucite tube lacks an introductory spiral. The fraction of air flowing out the cold pipe that corresponds to maximum temperature fall ranges from about 0.4 down to 0.25. The temperature fall increases with nozzle pressure, although not linearly. With Hilsch's tubes, the larger the tube, the greater the temperature fall.

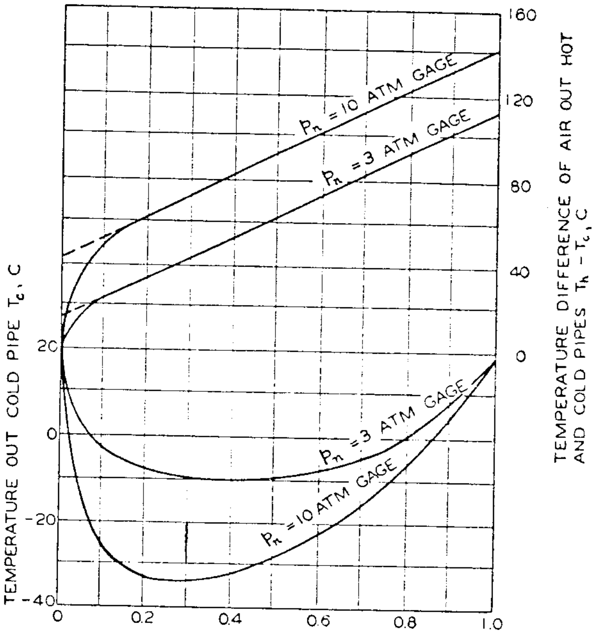


Fig. 3. Cooling as a function of air ratio. (Hilsch tube, model l-c, data from ref. 2. Nozzle temperature $T_n = 20$ C.).

Figure 3 (Hilsch's data) shows how cooling varies with fraction of flow through the cold pipe—that is, how the point of minimum temperature is approached as the fraction of air flowing out the cold pipe varies. The upper curves bring out the fact that, for the Hilsch l-c tube at least, the difference between the hot-pipe temperature and the cold-pipe temperature varies linearly with the fraction of air flowing out the cold pipe over most of the range.

The efficiency, or performance coefficient,* of the vortex-tube refrigerator is not high. The performance coefficient is defined as the ratio of heat abstracted to work expended[5] and, for a home refrigerator using Freon refrigerant, is around 4. The minimum work for constant-temperature compression of a mole of ideal gas from atmospheric to nozzle pressure is

$$A = RT_n \ln \frac{p_n}{p_{atm}}$$

The amount of cooling produced by the vortex tube per mole of air through the nozzle is

$$Q = \mu c_p M (T_n - T_c)$$

For the Hilsch l-c tube using air at 5 atm abs, $(T_n - T_c)_{max}$ is 38 C, and μ^* is 0.38; consequently, since c_p is nearly constant over this pressure range, $Q = (0.38)\ (0.24)\ (29)\ (38) = 101$ C.c.u. per lb mole The corresponding amount of minimum compression work is $A = (1.98)\ (293)\ ln5 = 935$ C.c.u. per lb mole. Therefore, the performance coefficient is

$$B = 101/935 = 0.11$$

This is only 1/40 as good as the vapor-compression refrigerator.

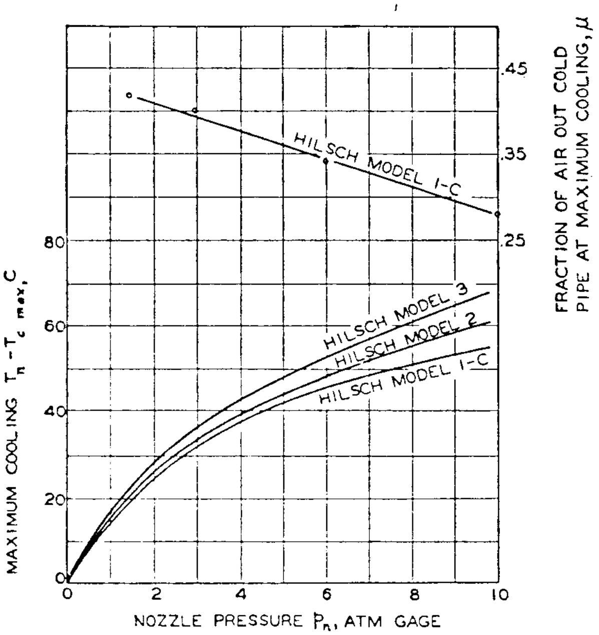


Fig. 4. Cooling as a function of nozzle pressure. (Nozzle temperature, T_n = 20 C. Data from Hilsch model of ref. 2.).

NOMENCLATURE

a	acceleration (centrifugal), ft per sec^2
A	minimum work of compression, ft-lb
c	velocity of sound at some specified temperature, ft per sec
c_p	heat capacity at constant pressure, ft-lb$_{force}$ per °C, lb$_{mass}$ 338$_{for\ air}$
c_p''	heat capacity at constant pressure, C.h.u. per °C, lb$_{mass}$; or Btu per °F, lb, 0.24$_{for\ air}$
D	diameter of the object indicated by subscript, ft
E	energy, ft-lb
F	force, lb
g_c	force conversion constant, 32.2 ft-lb$_{mass}$ per lb$_{force}$ sec^2
k	ratio specific heats, dimensionless
k_1	proportionality constant used in derivation: 1/sec
K	arbitrary constants
L	length, ft
m	mass, lb
M	molecular weight (29 for air)
n	exponential constant used in a derivation, dimensionless
p	pressure, lb force per sq ft
Q	quantity of heat, C.h.u.
r	radius of the object specified by subscript, ft
R	gas constant 2780 ft-lb per °C., lb$_{mole}$
S	cross-sectional area, sq ft
t	time, sec
T	temperature, usually K
V	velocity of rotating gas at point specified by subscript, ft per sec
W	weight rate of discharge of orifice or nozzle, lb per sec
Z	$(r_o/r_w)^2$, ratio of orifice area to cross section of tube, dimensionless
α	angle between flow streamline and circle based on center of rotation
ρ	density, lb mass per cu ft
β	refrigerator performance coefficient, dimensionless
μ	fraction of gas flowing out cold pipe, dimensionless
Subscripts	
a	axis of rotation
c	cold pipe
h	hot pipe
n	nozzle
o	orifice
w	wall of vortex tube

Hypothesis of Operation

In the Hilsch tube, gas is introduced tangentially at high velocity through a nozzle. Consequently, there is a gas mass present in the tube, rotating rapidly. A portion of the total flow entering the tube is removed at the axis of rotation through a hole in the diaphragm that closes that end of the tube (the diaphragm butts against the side of the nozzle); this inward flow proceeds along a spiral path. The remainder of the gas leaves the entrance region by whirling off down the wall toward the throttling valve. The nozzle, orifice, and spiral path are in essentially the same plane, which is parallel and adjacent to the diaphragm. Cold gas leaves through the orifice, hot gas through the throttling valve.

One of the most noticeable facts about data on the Hilsch tube is the wide spread between hot and cold stream temperatures that can be obtained under certain conditions. Thus, Figure 3 shows a difference as great as 140 deg C (about 250 deg F) when the operating conditions are such that most of the air is leaving the cold pipe and only a small flow of very hot air is leaving the hot pipe. Thus, a hypothesis on the Hilsch tube must explain not only the cooling of the one stream, but also the mechanism by which the energy abstracted from this cold stream is added to the portion of gas that becomes the hot stream. An energy balance indicates that such exchange is required, but does not show how it is done.

The view taken in this paper is that energy transfer outward from any given point in the whirling mass occurs as a sort of recoil reaction to the inward expansion of the gas at that point. Since the rigid wall does not permit work to be done outward along a radius (that is, there is no "give" in this direction, or no distance that the force acts through), the work must be done tangentially and consists of a forward push on that gas element which is just outside the element in question. The ultimate resisting force is the friction of the rotating gas mass on the wall, which, it should be noted, is tangentially directed. The possibility of this forward component of recoil becomes more apparent when it is recalled that, because of the spiral nature of the flow, the streamlines of that portion of gas moving toward the axis are not concentric with the wall but instead form an angle with the wall at any point. This tangentially directed (but outwardly diffusing) work is taken out of the enthalpy store of the gas through the mechanism of expansion and causes an increase in the rotation rate of the mass that is a function of the fraction of gas moving in along the spiral (that is, the fraction that the cold stream represents). This tangential work is eventually converted to heat by friction at the wall and at the throttling valve, and the heat is carried off in the hot stream.

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* Conditions at time of, or related to, minimum temperature in cold pipe.

As part of the same process, there is an inward radial expansion against the centrifugal field. This expansion work is also taken out of the enthalpy store of the gas. By postulating that work is done by (that is, energy transferred from) the gas in a stream tube only in a direction perpendicular to the stream tube, it is possible to show that the tangential and radial components of the expansion work are equal and, furthermore, that this equality results in a condition of constant angular velocity in the rotating mass. Since the angular velocity is constant, it can be characterized by the linear velocity at the wall (and the tube radius). It is of interest to note that for Hilsch's l-c tube operating on air at 10 atm gage, and using the concepts developed in this paper, the velocity at the wall when none of the gas leaves through the cold pipe is about 800 ft per sec; the difference between this value and the approximate 1000 ft per sec velocity that the gas has when it enters the tube from the nozzle is caused by wall friction. On the other hand, when all the gas is forced to move inward along the spiral path, causing a maximum of forward recoil thrust, the wall velocity is about 1400 ft per sec; this also happens to be the velocity of sound in air at the temperature of air at the wall.

APPENDIX: Details of the Hypothesis

Equivalence of Radial and Tangential Expansion Work—Figure 2A is a representation of a cross section of the Hilsch tube near the orifice diaphragm. The spiral line is a streamline for that portion of the total flow that eventually leaves through the orifice. (A streamline is simply a line which shows the direction of the flow velocity at successive points in a flow field, when starting from a given point; a great number are required to describe the whole flow field. For viscous motion along a straight pipe, the streamlines are straight; for a "free" vortex, they are circles; for the case being examined, they are spirals.)

The rapid rotation causes a pressure gradient that increases from axis to wall; the pressure gradient is the physical, measurable, aspect of the centrifugal field accompanying the rotation. If no gas were being admitted at the periphery, a condition of equilibrium would prevail (neglecting friction) in which a given element of gas would continue to move in a circle of fixed radius; since this system is in equilibrium, no work would be done by any portion of the gas. However, when gas is admitted at the periphery and removed at the axis, it travels in a spiral path from a region of high pressure to a region of low pressure, and expands and cools in the process. From Figure 2A it is seen that at any point along the streamline, the streamline is making an angle, α, with a circle drawn through that point and centered on the axis. Consequently, expansion occurring perpendicular to the streamline can be resolved into two components, one being an inward expansion along a radius and against the centrifugal field, and the other being a forward expansion along a tangent and against (ultimately) the frictional drag on the wall. The force involved in the latter action, by its reciprocal nature, while tending to cause speeding of a gas element on the "forward," or outer, side of the streamline, simultaneously causes slowing of a gas element on the "back," or inner, side of the streamline, with the result that there is a decrease in tangential velocity with a decrease in radius. This change of velocity with radius is shown in the following paragraphs to be such as to produce a condition of constant angular velocity in the rotating mass. Also, the radial component of expansion work is shown to equal the tangential component of expansion work.

Figure 2B shows a very short length of spiral path, $V\mathrm{d}t$, for an element of gas moving in toward the axis; the length, being a differential segment, appears straight. The coordinate, Y, is a radius of the tube, **X is** a coordinate perpendicular to Y, and α is the angle between $V\mathrm{d}t$ and X. While moving the distance $V\mathrm{d}t$, the gas element, by composition of velocities, can also be considered to have moved inward a radial distance

$$\mathrm{d}l_r = V\mathrm{d}t \sin \alpha \tag{1}$$

and forward a tangential distance

$$\mathrm{d}l_T = V\mathrm{d}t \cos \alpha \tag{2}$$

In Figure 2C, two aspects of the force F acting at the gas element are shown: they are what was called by Newton "action and reaction." The "upward" aspect is able to push the element in toward the axis of rotation by the force component

$$F_r = F \cos \alpha \tag{3}$$

The "downward" aspect is able to push the adjacent gas element forward in the direction of rotation by means of the force component

$$F_T = F \sin \alpha \tag{4}$$

Since work is the product of a force times the distance through which it acts, the work done in a radial direction by the gas element in time $\mathrm{d}t$ is obtained from Equations 1 and 3 as

$$\mathrm{d}W_r = F_r \mathrm{d}l_r = V\mathrm{d}t\ F \sin \alpha \cos \alpha \tag{5}$$

Similarly, the work done in a tangential direction by the gas element in time $\mathrm{d}t$ is obtained from Equations 2 and 4 as

$$\mathrm{d}W_T = F_T \mathrm{d}l_T = V\mathrm{d}t\ F \sin \alpha \cos \alpha \tag{6}$$

Equations 5 and 6 show that expansion work done radially against the centrifugal field is equal to expansion work done tangentially against neighboring elements. It should be noted that the value of α—that is, the exact nature of the spiral path—has no bearing on this equivalence.

Constant Angular Velocity for Rotating Mass—There is only one way in which the linear velocity of rotation of this gas mass can vary with the radius of rotation and still maintain the equivalence between radial and tangential work, and that is to have the velocity proportional to the radius—a condition of constant angular velocity. This is arrived at as follows. Assume the law of variation to be

$$V = k_1\ r^n \tag{7}$$

The equation for centrifugal force acting on mass m is

$$F = \frac{mV^2}{g_c r} \tag{8}$$

or, with Equation 7,

$$F_r = \frac{mk_1^2\ r^{2n-1}}{g_c} \tag{9}$$

The work done against this force and through distance $\mathrm{d}r$ is

$$\mathrm{d}W_r = F_r\ \mathrm{d}r = \frac{mk_1^2}{g_c} r^{2n-1}\mathrm{d}r \tag{10}$$

The tangential work done is equivalent to the change in kinetic energy that corresponds to the change in linear velocity with radius. The kinetic energy of a mass m moving with velocity V is

$$E = m\frac{V^2}{2g_c} = \frac{m}{2g_c} k_1^2\ r^{2n} \tag{11}$$

(by use of Eq. 7). The change in kinetic energy with radius is obtained by differentiating Equation 11:

$$\mathrm{d}E = \mathrm{d}W_T = 2n\frac{m}{2g_c} k_1^2\ r^{2n-1}\ \mathrm{d}r \tag{12}$$

When Equations 10 and 12 are equated, according to the principle arrived at in the last paragraph, it is found that $n = 1$, and consequently that

$$V = k_1 r \tag{7a}$$

Thus, a condition of constant angular velocity prevails in this rotating gas mass. If V_W is the velocity at the tube wall of radius r_W, it is obvious that

$$k_1 = V/r = V_W/r_W \tag{7b}$$

This relation will be used later.

Although a condition of constant angular velocity for the gas mass in this tube appears to be at odds with the observed velocity distribution in a cyclone collector, which lies between the extremes given by constant angular velocity and constant angular momentum, the difference is believed to be due to the extremely small amount of expansion work done by gas in the cyclone. The gas in a cyclone is, for our purposes, an incompressible fluid and should obey the law of constant angular *momentum* ($V = k/r$), except as influenced by viscosity and turbulence.

Values of Radial and Tangential Expansion Work—By inserting the foregoing values for n and k_1 into Equation 10, and taking the mass m as unity, we obtain

$$\mathrm{d}W_r = \left(\frac{V_W}{r_W}\right)^2 \frac{r\ \mathrm{d}r}{g_c} \tag{10a}$$

as the work done on unit mass, against the centrifugal field, through the distance $\mathrm{d}r$. Integrate between $r = 0$ and $r = r$ to obtain

$$W_r = \left(\frac{V_W}{r_W}\right)^2 \frac{r^2}{2g_c} \tag{13}$$

This is the work done on one pound of gas by the centrifugal field in going from the axis to any radius r ("downhill," in effect). The negative of this is the work done on the gas by the field in going in the opposite direction, namely from any radius to the axis ("uphill"). From the principle derived previously, the tangential work is the same, so that the *total* work done on one pound of gas in going from any radius to the axis is twice Equation 13, or

$$W = \left(\frac{V_W}{r_W}\right)^2 \frac{r^2}{g_c} \tag{13a}$$

It is readily seen that the work done *on* the gas between r_1 and r_2 is

$$W_{1\text{-}2} = (r_2^2 - r_1^2)\ \frac{1}{g_c}\left(\frac{V_W}{r_W}\right)^2 \tag{14}$$

Energy Balance in Separating Zone—We can now set up an energy balance along a streamline. It will differ in one term from the usual energy balance: the field potential term h, denoting the energy that can be recovered from one pound of matter a distance h above a datum plane in a gravitational field, will have to be replaced by C, the energy that can be recovered from one pound of matter because of its position in a *centrifugal* field. From the definition of potential fields, the change in potential is equal to the work required to move unit mass from one place to another against the field. Equation 13 gives the magnitude of this work in going from a radius r to the axis. If we take the axis as our datum level, then all other points are at a lower potential and we have to use the negative sign

$$C = -\left(\frac{V_W}{r_W}\right)^2 \frac{r^2}{2g_c} \tag{15}$$

The complete energy balance is then.

$$C_1 + p_1v_1 + E_1 + \frac{V_1^2}{2g_c} + W_{1\text{-}2} = C_2 + p_2v_2 + E_2 + \frac{V_2^2}{2g_c} \qquad (16)$$

The term for heat transport, Q, into the element has been dropped out because subsequent calculations indicate that it is of minor importance. If the pv, and E (internal energy) terms are lumped together into the usual enthalpy term H, and values substituted for C (Eq. 15), $W_{1\text{-}2}$ (Eq. 14), and V^2 (Eq. 7b), we obtain

$$-\frac{r_1^2}{2g_c}\left(\frac{V_W}{r_W}\right)^2 + H_1 + \frac{r_1^2}{2g_c}\left(\frac{V_W}{r_W}\right)^2 + \frac{(r_2^2 - r_1^2)}{g_c}\left(\frac{V_W}{r_W}\right)^2 = -\frac{r_2^2}{2g_c}\left(\frac{V_W}{r_W}\right)^2 + H_2 + \frac{r_2^2}{2g_c}\left(\frac{V_W}{r_W}\right)^2 \qquad (16a)$$

This reduces to

$$H_1 - H_2 = \frac{(r_1^2 - r_2^2)}{g_c}\left(\frac{V_W}{r_W}\right)^2 \qquad (17)$$

Since by definition

$$dH = C_p\ dt \qquad (18)$$

$$\text{then } H_1 - H_2 = C_p\ (T_1' - T_2')^* = \frac{(r_1^2 - r_2^2)}{g_c}\left(\frac{V_W}{r_W}\right)^2 \qquad (19)$$

Equation 19 gives us the information that if we were to remove gas from two different radii in the whirling mass and stagnate the two streams (to convert the kinetic energy into enthalpy), they would have a resulting temperature difference equal to *twice* their difference in kinetic energy (using C_p to convert to temperature terms). The general magnitude of this temperature difference checks with Hilsch's data (Fig. 3), as will be shown later. If r_2 is taken at the axis, its value is zero and Equation 19 becomes

$$T' - T_a' = \left(\frac{r}{r_W}\right)^2 \frac{V_W^2}{C_p\ g_c} \qquad (19a)$$

With the radius taken at the wall of the tube ($r = r_W$), the relation becomes

$$T_W' - T_a' = \frac{V_W^2}{C_p\ g_c} \qquad (19b)$$

Similarly, at a radius equal to the radius of the orifice

$$T_o' - T_a' = \left(\frac{r_o}{r_W}\right)^2 \frac{V_W^2}{C_p\ g_c} = Z\ \frac{V_W^2}{C_p\ g_c} \qquad (19c)$$

where Z is defined as the ratio of the orifice area to the tube cross section. It should be emphasized that the temperature differences in Equations 19 to 19c are values that would be measured after stagnating the gas concerned, as happens with the hot and cold streams leaving the Hilsch tube. The temperature difference indicated by instruments traveling *with* the gas (so as to eliminate the temperature rise associated with conversion of the kinetic energy) will be one-half the above values.

Pressure at Tube Wall in Separating Zone—Before proceeding further with analysis of temperature effects, it will be desirable to have a quantitative description of the pressure gradient existing in the "separation zone" of the tube. This pressure gradient is the physical aspect of the centrifugal field present. It can readily be shown that the pressure gradient due to the centrifugal field is

$$\frac{dp}{dr} = \frac{V^2}{r}\ \frac{\rho}{g_c} \qquad (20)$$

Since the density at any point is MP/RT, and the velocity at any point in a mass rotating with constant angular velocity is $r\ V_W/r_W$, Equation 20 becomes

$$\frac{dp}{dr} = \frac{r}{g_c}\left(\frac{V_W}{r_W}\right)^2 \frac{Mp}{RT} \qquad (20a)$$

If an "average" value of temperature throughout the mass is used, Equation 20a integrates to

$$ln\ \frac{p}{p_a} = r^2\ \frac{M}{2g_cRT}\left(\frac{V_W}{r_W}\right)^2 \qquad (21)$$

Equation 21 gives numerical values that are fairly close to those obtained by Equation 22, below, which indicates that a precise knowledge of the temperature throughout the rotating mass is of second order importance as far as pressure is concerned. However, this refinement can be made by using in Equation 20a the adiabatic relation between temperature and pressure, as follows.

The adiabatic relation requires a constant entropy process, which means no change in internal energy through either heat transfer or frictional degradation of mechanical energy. It is obvious that this condition cannot be met exactly, but it is also apparent that, since the experimental temperature falls in the Hilsch tube are so large, the heat transfer and mechanical energy degradation cannot be of major importance. It can be seen that the adiabatic relation holds roughly, at least, for, as the gas moves toward the center, both the pressure and temperature decrease. The adiabatic relation is

$$\frac{T}{T_a} = \frac{(p)}{(p_a)}^{k-1/k}$$

When T in Equation 20a is replaced by its equivalent from this relation, Equation 20a can be integrated to

$$\frac{p}{p_a} = \left[1 + \left(\frac{r}{r_W}\right)^2 \frac{V_{W^2}}{2g_cC_p\ T_a}\right]^{k/k-1} \qquad (22)$$

By use of the adiabatic relation, it follows that

$$\frac{T}{T_a} = 1 + \left(\frac{r}{r_W}\right)^2 \frac{V_W^2}{2g_cC_p\ T_a} \qquad (23)$$

or,

$$T - T_a = \left(\frac{r}{r_W}\right)^2 \frac{V_W^2}{2g_cC_p} \qquad (23a)$$

It will be seen that Equation 23a is one-half the value given by Equation 19a. However, if the gas removed from point r is stagnated, to convert the kinetic energy back into enthalpy the subsequent temperature difference, $T' - T_a'$, equals that given by Equation 19a. All that the agreement means is that the same temperature relation is arrived at, whether approached through the potential energy aspect (Eq. 19a) or the force aspect (Eq. 23a) of the centrifugal field.

A short experiment was performed to test the general correctness of the pressure equation (Eq. 22). A hypodermic needle with point beveled at about 30 deg from its axis was found to serve fairly well as an impact tube when inserted into a moving stream, with the axis perpendicular to the stream and the opening facing upstream. For example, when the upstream pressure on a critical flow nozzle read 30 psig, the pressure indicated by a gage connected to the needle inserted perpendicularly into the jet was 28 psig. The impact tube was calibrated in this fashion for use in the following experiment. A small hole was drilled in the wall of a Lucite vortex tube opposite the point of entry of the nozzle. By connecting a pressure gage to the hole, the static pressure at the wall could be measured; and by inserting the hypodermic needle, the total pressure (impact as well as static pressure) at wall and axis could be determined. With a nozzle pressure of 50 psig, it was observed that the "static" wall pressure was 15 psig, total pressure at the axis was 1 psig, and total pressure at the wall (corrected for measuring error) was 29 psig. Actually, because the pressure-tap hole was in a curved wall, it tended to scoop into the rotating mass and pick up a fraction of the velocity pressure, calculated to be around 1/10 (hole diameter about 1/6 of tube diameter). With this correction, the observed static pressure at the wall becomes 13.5 psig, and the observed velocity head becomes 15.3 psig (13.5 + 15.5 = 29). These data can be used to evaluate Equation 22 (it is found that V_W is close to 1000 ft per sec), which then indicates a static wall pressure of 12.5 psig. This is in good agreement with the observed value of 13.5 psig, in view of the difficulty in making accurate pressure observations. The main point that the experiment corroborates is that of constant angular velocity; it does not prove much about temperatures since, as previously mentioned, the centrifugal pressures are not particularly dependent on accurate knowledge of the temperature gradient.

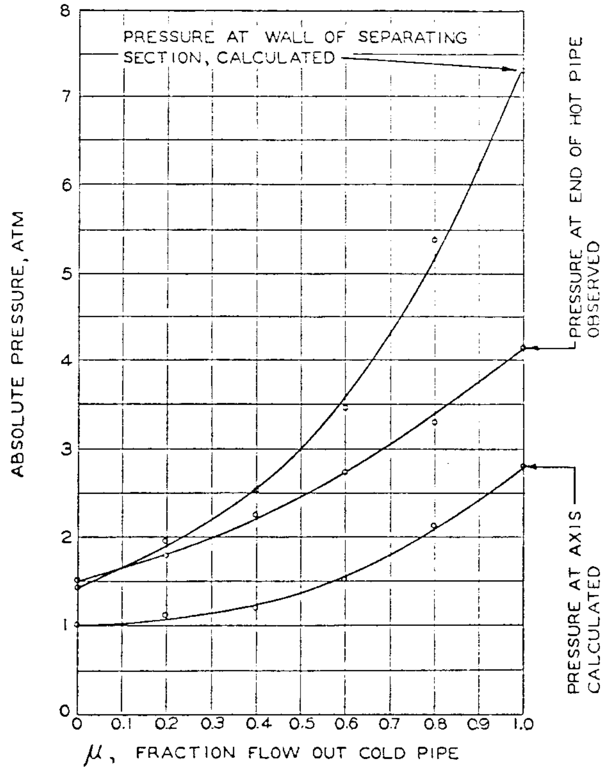

Fig. 5. Rotary velocity at wall of Hilsch model 1-c tube. (Nozzle pressure at 10 atm gage.)

Pressure at Axis in Separating Zone—Inasmuch as pressure at any point in the active region depends on pressure at the axis, according to Equation 22, it is of interest to know the pressure at the axis. The latter is dependent essentially on the cross-sectional area of the orifice and on the amount of gas that has to pass through the orifice. The amount of gas entering the tube through the nozzle is given by Fliegner's equation for critical flow of air, modified for deg K in place of deg R, and using a coefficient of unity:

$$W_n = 0.397\ S_np_n\ \frac{1}{\sqrt{T_n}}\ \text{lb per sec} \qquad (24)$$

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* ' (prime) Temperature or pressure that includes both static and kinetic components of rotating gas.

Since the fraction of total gas that leaves through the orifice is defined to be μ, the weight of gas through the cold pipe is

$$W_c = \mu \, 0.397 \, S_n p_n \frac{1}{\sqrt{T_n}} \tag{24a}$$

In the happy event that flow through the orifice also follows the critical flow relation (requires pressure on the inside face of the orifice to be above 1 atm, gage), we can write

$$W_c = 0.397 \, S_o p_{avg} \frac{1}{\sqrt{T_a}} \tag{24b}$$

By elimination between Equations 24a and 24b, it is apparent that

$$p_{avg} = p_n \mu \frac{S_n}{S_o} \sqrt{\frac{T_a}{T_n}} = p_n \mu \left(\frac{D_n}{D_o} \right)^2 \sqrt{\frac{T_a}{T_n}} \tag{25}$$

The complexities of this particular flow situation became evident here: p_{avg} is the average pressure across the inside face of the orifice, used simply because an exact statement in terms of the centrifugal pressure gradient is too complex to be usable; however, error from this source probably does not exceed 10 percent of the absolute pressure at the axis. Another point of confusion is the degree of contraction of the gas stream as it passes through the orifice: in Equation 24b a coefficient of unity for the orifice was used—that is, it was assumed that no contraction would occur, the tendency being offset by the centrifugal acceleration, which has values well over 100,000 "*g's*" even at the radius of the orifice; nevertheless, the validity of this assumption is not clearcut.

If pressure at the inside face of the orifice is too low to cause critical flow, the cumbersome equation for adiabatic expansion through an orifice may be used to determine the pressure drop. At these low values, though, it is probably just as satisfactory to neglect compressibility effects and use the orifice equation for incompressible fluids, which for this situation reduces to

$$p_{avg} - p_{atm} = \frac{W_c^2}{S_o^2 \rho 2 g_c} \tag{26}$$

The density, ρ, is the arithmetic average density of the gas during its expansion from p_{avg} to the atmospheric pressure which exists in the cold pipe. We have again assumed a coefficient of unity for rotational flow through the orifice. The value of w_c is obtained by Equation 24a.

Figure 5 gives plots of pressure at different points in the Hilsch 1-c tube at a nozzle pressure of 10 atm gage, with the flow ratio as the independent variable. Pressures at the axis in the separating sections were obtained by use of Equation 26 when flow through the orifice was below critical, and by use of Equation 25 when flow through the orifice was critical. Pressures in the hot pipe, just ahead of the valve, were taken from Hilsch's reported values. Pressures at the wall of the separating section were calculated with Equation 22; the necessary values for V_W came from curve C, Figure 6, based on Hilsch's measured temperatures, as described later.

It is interesting to note the relative magnitudes of these pressures. Flow from the wall region in the separating zone down through the hot pipe corresponds to the direction of the pressure gradient between these two points. For the same reason, it is expected that there is flow, or recirculation, from the hot pipe back into the lower-pressure region at the axis of the separating zone. The recirculation would tend to decrease the temperature difference between hot and cold streams, and such an apparent effect is noticed in the data given in Figure 3, where there are abnormally low values for $T_h - T_c$ at low values of μ (where the rate of "energy separation" is low). In addition, it is probably true that any condition which interferes with removing the hot stream from the separating zone as smoothly as possible will decrease the temperature separation. For example, a short hot pipe, a large diameter hot pipe, or a straightening vane in the hot pipe would all tend to cause a pressure recovery in the hot stream at a point close to the separating zone, with consequent increase in recirculation back to the low pressure area at the axis of the zone.

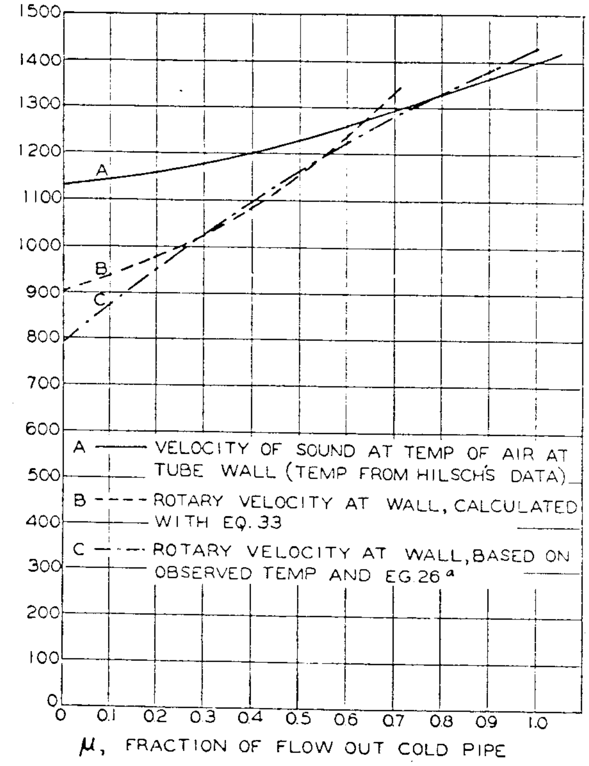

Fig. 6. Pressures in Hilsch model 1-c tube. (Nozzle pressure at 10 atm gage.)

Temperature Difference Between Gas Out of Hot and Cold Pipes—The foregoing analysis of pressure in the active, or separating, section of the tube is of aid in the process of determining the *average* temperature of gas out of the hot and cold pipes. It is obvious, for example, that the temperature of the gas flowing out the cold pipe lies somewhere between gas temperature at the axis and gas temperature at a radius equal to the orifice radius. A rigorous derivation of a general relation for the average temperature was not possible with our mathematics, but a fair approximation was obtained by numerical evaluation of several representative cases. It appears that a reasonable compromise is obtained by considering that the temperature of the cold stream, after mixing and stagnation, is equal to the stagnated temperature of gas taken from a point at which the radius is 0.8 of the orifice radius, or $(r/r_o)^2 = 0.64$ (will call 0.65). The general reason for this can be seen by the fact that if the pressure were uniform at all radii, the representative radius would be that of a circle with one-half the orifice area, or a radius 0.707 of the orifice radius. The pressure gradient, however, biases the representative point outward to the vicinity indicated; the bias comes about through increase in density with radius, and, in subcritical flow, through increase in velocity through the orifice. Consequently, the temperature of gas out the cold pipe is

$$T_c - T_a = 0.65 \left(\frac{r}{r_W} \right)^2 \frac{V_W^2}{C_p \, g_c} = 0.65Z \frac{V_W^2}{C_p \, g_c} \tag{24}$$

The error possible in $T_c - T_a$, because of error caused by use of a fixed factor (0.65) instead of a factor that shifts with conditions, is only a few degrees and does not invalidate the purpose of this section of the paper, which is to indicate the trends in performance caused by changes in the variables of construction and operation.

Determination of a general relation for the average temperature for gas out of the hot pipe presents even more difficulties than the above case for flow out of the cold pipe. The additional complication is that the inner boundary for flow is not fixed, but changes with flow ratio and other operating and structural variables. The reason for this is that the gas will not move from a given point in the rotating mass in the separating zone down the hot pipe unless pressure at that point is greater than pressure in the hot pipe. The point, or radius, of pressure equality tends to shift out toward the wall as values of μ decrease. At a nozzle pressure of 10 atm gage in the Hilsch 1-c tube, the radius of pressure equality—that is, the inner boundary of the moving annulus—is around 0.7 of the wall radius when $\mu = 0.5$; thus, in this case, all gas leaving through the hot pipe has come from regions at higher temperatures and higher pressures than found at 0.7 r_W. Again, to simplify the problem, a compromise was reached on a representative radius: it was decided to consider gas from a point at 0.95 r_W as representative generally of gas out of the hot pipe. Since at this point $(r/r_W)^2 = 0.9$,

$$T_h - T_a = 0.9 \, (T_W' - T_a') = 0.9 \frac{V_W^2}{C_p \, g_c} \tag{25}$$

From Equations 24 and 25, it is seen that

$$T_h - T_c = (0.9 - 0.65Z) \, (T_W' - T_a') = (0.9 - 0.65Z) \frac{V_W^2}{C_p \, g_c} \tag{26}$$

For the Hilsch 1-c tube $Z = 0.23$, so

$$T_h - T_c = 0.75 \frac{V_W^2}{C_p \, g_c} \tag{26a}$$

Actual Temperature of Hot and Cold Streams—So far, we have related hot and cold stream temperatures but have not shown how they are related to the temperature of the compressed air we started with. This relation is obtained with an energy balance on the one stream in and the two streams out of the device. However, since the gravitational energy terms cancel and the kinetic energy terms are insignificant, the general energy balance reduces to an enthalpy, or heat, balance. Call the fraction of total flow that goes through the cold pipe μ, and take the specific heat of all streams as being the same, then the balance reduces to

$$T_n = \mu T_c + (1 - \mu) \, T_h \tag{27}$$

Subtract T_c from both sides and obtain

$$T_n - T_c = (1 - \mu) \, (T_h - T_c) = (1 - \mu) \, (0.9 - 0.65Z) \frac{V_W^2}{C_p \, g_c} \tag{28}$$

The apparent conflict between Equations 26 and 28—namely, that for a constant value of V_w, $T_h - T_c$ is independent of the flow ratio (μ) while $T_n - T_c$ is not—is resolved when it is realized that the temperature level of the whole rotating mass shifts upward or downward around T_n, depending on the flow ratio. To take the extremes, when $\mu = 1.0$, all the gas leaves through the cold pipe at T_n, but the temperature of the rotating mass still inside the tube is greater than T_n; conversely, with $\mu = 0$, all the gas leaves through the hot pipe at T_n, but the temperature of the rotating mass is now below T_n. In either case, according to the equations, as long as V_w remains fixed, the temperature extremes in the tube are fixed. Actually, the recirculation mechanism discussed earlier causes some degree of deviation from this relation.

Magnitude of Rotary Velocity at Wall—The remaining major point to be evaluated is the manner in which V_w, which appears in most of the equations, changes with the operational and structural variables of the tube. V_w is considered to take on a value at which the rate of energy loss resulting from friction against the wall is just offset by the rate of gain from the jet plus the rate at which mechanical energy is transferred back from the spiral. The kinetic energy rate of the jet is the product of pounds per sec of gas times the kinetic energy of a pound of gas, or $W_n V_n^2/2g_c$. With Equation 24 to give the value of W_n, we obtain

$$(E/t)_{jet} = 0.397 \frac{\pi D_n^2 p_n}{4 \sqrt{T_n}} \frac{V_n^2}{2g_c} \tag{29}$$

It is quite likely that we can assume without too much error that V_n is essentially constant at the sonic velocity with which it leaves the jet (1030 ft per sec at $T_n = 20$ C); in any event, the contribution to the forward rotation, made by additional expansion of the jet at the wall of the tube, cannot readily be calculated since shock waves caused by the curved wall will probably prevent much velocity increase of gas in the jet.

The amount of energy converted from its form as enthalpy of the rotating mass into a tangentially directed mechanical form at the tube wall is, by the principle culminating in Equation 13a, $V_w^2/2g_c$ ft-lb per lb of gas. For a gas rate of μW_n moving through the spiral, this becomes

$$(E/t)_{spiral} = 0.397 \mu \frac{\pi D_n^2 p_n}{4 \sqrt{T_n}} \frac{V_W^2}{2g_c} \tag{30}$$

The rate of energy input sustaining forward rotation is the sum of Equations 29 and 30:

$$(E/t)_{input} = W_n \left(\frac{V_n^2}{2g_c} + \frac{V_W^2}{2g_c} \right)$$

$$= 0.397 \frac{\pi D_n^2 p_n}{4 \sqrt{T_n}} \left(\frac{V_n^2}{2g_c} + \frac{V_W^2}{2g_c} \right) \tag{31}$$

The rate of energy dissipation by the rotating mass is of a more speculative nature, but it seems possible to approximate it. Hilsch's data on the 1-c tube are best fitted by the assumption that the drag force is proportional to the wall area against which the "separating zone" is rotating, to the first power of the velocity at the wall, and to a fixed "friction coefficient." If we assume the length of the separating zone to be proportional to the diameter of the inlet nozzle, and remembering that force times velocity gives an energy rate, then

$$(E/t)_{friction} = f \pi D_W D_n \frac{V_W^2}{2g_c} \tag{32}$$

Here f must have the dimensions of lb mass/sec $(ft)^2$ to balance the equation. It was initially expected that this friction relation would involve a velocity head, similar to the friction relation for turbulent flow in pipes, but data on the Hilsch 1-c tube indicate that the drag force is proportional to neither the gas density nor the second power of the velocity. It resembles, in short, a viscous drag, but reasons for this are not clear.

When Equations 31 and 32 are equated and similar terms canceled, we obtain

$$\left(\frac{V_n}{V_W} \right)^2 = \frac{4f D_W \sqrt{T_n}}{0.397 D_n p_n} - \mu \tag{33}$$

The unknown constant f can be evaluated from Hilsch's data, reproduced in Figure 3. Here $T_h - T_c$ is given as a function of the flow ratio, μ, for Hilsch's tube 1-c at $p_n = 10$ atm gage. From Equation 26a, V_w can be obtained directly; the errors due to ignoring recirculation and to the approximations used to obtain average temperatures out hot and cold pipes, will be incorporated, of course. Curve C on Figure 6 is a plot of V_w vs μ for the Hilsch 1-c tube at $p_n = 10$ atm gage. Data at $\mu = 0.3$, the point of maximum temperature fall, were used to evaluate from Equation 33. The resulting value was $f = 40$ lb/ sec $(ft)^2$. With the magnitude of the friction coefficient established, Equation 33 was used to calculate the plot of V_w vs μ given as curve B on Figure 6. The agreement between theoretical and empirical curves is fair in the middle section. The empirical curve is believed to be too low at low values of μ because of recirculation effects (depress $T_h - T_c$ on which V_w is based). The theoretical curve is too high at high values of μ because it ignores the limitation set by the speed of sound in gas at the wall. That is, shock waves will be formed when the gas velocity at the wall tends to exceed sonic velocity and further velocity increase will be prevented. Line A on Figure 6 is a plot of this limiting velocity, based on Hilsch's hot-stream temperatures and a table of sound velocity in air as a function of temperature. It can be seen that at high velocities it falls fairly close to the "observed" velocities given by curve C of Figure 6.

The analysis of motion ending in Equation 33 is not satisfactory for several reasons: it assumes constant jet velocity; it does not account for the energy lost in turbulence when the jet shoots into the rotating mass; and it considers the width of the frictional area to the fixed (proportional to D_n), when it very likely changes with flow rate and other variables. However, further theoretical work probably is not justified until more specific experimental data on motion in the tube are obtained by groups interested in the commercial possibilities of the tube. It is hoped that the viewpoint presented in this paper will be of value to any such investigators.

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Discussion

Neil P. Bailey, Rensselaer Polytechnic Institute

Equation 14 of this paper is the well known first approximation for the work done on the impeller of a centrifugal turbine. Mr. Webster has not realized that he has written the equations for radial and tangential decelerations that can be true *only* when the flow is passing through an impeller upon which the air does work. This same equation, (14), applies equally well for the work done by a radial blade centrifugal compressor impeller as air flows outward.

These equations simply do not apply to flow with no impeller because there is no mechanism present to produce such decelerations. Mr. Webster was further misled by the pure coincidence that the pressure and temperature changes in such a centrifugal impeller just happen to be comparable in magnitude to the measured pressure and temperature changes in a Hilsch vortex.

The following is a summary of the conclusions reached to date from studies on a free vortex at Rensselaer.

1) The ideal free vortex of the mathematician (velocity × radius = constant) cannot exist in a real gas that has the three "transport" properties of diffusion, viscosity, and thermal conduction.
    a. It calls for radial velocity gradients that are very large and cannot exist in a gas with viscosity.
    b. It calls for radial static temperature gradients that would result in high rates of radial heat transfer.
    c. It furnishes no possible rational solution near the center when an actual gas is considered.
2) It has been concluded analytically, experimentally, and philosophically that complete equilibrium with a real gas demands rotation at constant angular velocity (tangential velocity = constant × radius) with a constant static (not total) temperature throughout. This means that:
    a. The total temperature cannot be constant but must increase from the center to the outside. This equilibrium is reached by radial heat transfer and is the reason for the low center temperature and the high periphery temperature in the Hilsch tube.
    b. It means that equilibrium is reached only after there is no longer relative tangential motion between successive gas layers. For this reason the angular momentum per unit mass of gas is not constant, but has been redistributed by forces of viscous origin.
3) By assuming this type of vortex, the velocity, total temperature, and pressure gradients have been calculated on the basis of
    a. Constant angular velocity
    b. Constant static temperature
    c. Integrated enthalpy per unit length of vortex equal to that supplied by the gas that formed it
    d. Integrated angular momentum per unit length of vortex equal to the angular momentum of the supplied air that formed it
4) Experimentally, such vortices have been formed in tubes from one to four in. in diameter by a set of tangential vanes at the circumference of the closed end of the tube. As the spinning air progresses toward the open or discharge end of the tube, it has been repeatedly traversed and measured and the following results are consistently reproducible.
    a. At first the total temperature is constant throughout the air and the velocity at the periphery is greater than that called for by section (2).
    b. A point is reached further down the tube where the angular velocity is essentially constant. At this point, radial pressure, total temperature, and tangential velocity as measured check well with the theoretical results of section (3).

   c. As the gas travels on out the tube, wall friction destroys its angular momentum and the tangential velocity becomes low at the pipe wall first and this degradation progresses gradually to the center. Heat transfer and viscous forces gradually redistribute the energy until the total temperature is again constant and equal to the original value.

5) There are still some points about which we are not clear and work is currently in progress to help settle them.
   a. What minimum elapse of time is required for equilibrium to be reached?
   b. Can there be a transition from supersonic to subsonic tangential velocity in such a vortex?
   c. Why does such a vortex in free space (a whirlwind) assume a characteristic funnel or conical shape?
   d. Is there a characteristic "core size" for a given amount of angular momentum per pound of air? Cyclone cores suggest this.

Alexander Goetz, California Institute of Technology

The author has found an ingenious approach to the problem of treating thermodynamically some of the problems presented by the performance of the Hilsch vortex tube. It appears, however, problematical whether this or similar approaches will succeed in providing the mechanism by which the entering group of gas molecules is "split" into two groups of different energy levels so that the energy missing in one is added to the other. In the treatment of transfer phenomena, thermodynamics will have to assume the existence of at least a "steady" state (if not of a quasi-equilibrium) in a continuum. If flow velocity, dimensions, etc., are considered for this case from a gas-kinetic point of view, it is apparent that the time of "sojourn" of a molecule in the separation chamber of the Hilsch tube (10^{-3} to 10^{-4} sec) is small compared to the average interval between molecular collisions. Hence, the mechanism for an intermolecular energy exchange is absent at this stage.

The distribution of energy among the molecules immediately after leaving the nozzle must be quite heterogeneous (although quantitatively different from a Maxwell-Boltzmann distribution). Under these conditions it would appear permissible to assume that the molecules with high energy (speed) will preferentially contribute to the formation of the vortex while the low-energy molecules will stay behind and be forced into the center of the tube opposite the orifice.

A radial temperature gradient is thus established with a maximum near the orifice and gradually vanishing along the axis of the hot tube at a rate at which an increasing possibility for energy exchange will take place. It appears that such a mechanism will not have to be contradictory, but rather supplements the thermodynamic treatment of the radial and tangential momenta involved in the procession of the vortex along the hot tube. The very critical dimension of the diameter of the orifice, the importance of it being as close as possible to the nozzle, the condition of the exact rotational symmetry of the spiral, and, finally, the necessity for minimizing frictional losses (polishing of the hot tube) seem to support the assumption that the initial separation is due to a molecular kinetic phenomenon.

This suggested point of view does not alter the overall thermodynamic energy balance; it only postulates the "a priori" existence of hot and cold molecules for which the mechanism of the Hilsch tube provides only a peculiar mode of separation, possibly not dissimilar to the function of the Maxwellian "demon." The work required for such a separation is supplied externally by the compressor.

C. D. Fulton Jr., Mass. Institute of Technology

In Equations 1 through 4, the force is assumed to be perpendicular to the velocity. Thus the work is zero and the speed is constant. Since this cannot, of course, be assumed, it is necessary to replace the angle α in Equations 3 and 4 by another angle β.

If a particle is whirled on the end of a string and the string is pulled through a grommet, the angular momentum of the particle will be preserved. Thus the tangential velocity will be inversely proportional to the radius and the angular velocity will be inversely proportional to the square of the radius. The force will be in the radial direction at every instant and there will be no tangential force and no tangential work. The tangential work in circular motion, therefore, is not measured by changes in V, but by changes in Vr. It can be shown that Equation 12 should be written

$$dW_T = \frac{m}{g_c} (n + 1)\ k^2_1\ r^{2n-1}\ dr$$

If the vortex is free, having constant angular momentum, $n = -1$. It is seen that the equation is correct in giving zero tangential work for this case. The kinetic energy comes from the pulling of the string—i.e., from the centrifugal work, which emerges as energy in the tangential direction. If the vortex is forced, having constant angular velocity, $n = 1$. For this case the equation gives $dW = 2\ m/g\ k^2\ r\ dr$, $W = m/g\ k^2r^2 = m/g\ V^2$. This is twice the kinetic energy, and it shows that half of the tangential work goes into the kinetic energy and the other half goes into work against the centrifugal force. This is another example of how, in vortices, work applied in one direction may emerge in the other direction. The Newtonian rules of Cartesian systems cannot be applied to polar systems without proper transformation.

In circular flow, no tangential force can exist on a gas element except through the presence of blades, shearing friction, magnetic force, or the like because the pressure behind the element is the same as the pressure in front of the element. There is also no such quantity as tangential expansion or tangential expansion work. If friction, blades, and body forces are not considered, the tangential work is zero and the vortex is free.

The general energy equation for steady flow, Equation 16, can only be applied to a control surface assumed to exist somewhere in the field of flow through which the gas flows. The work in this equation is shaft work crossing the control surface. Under the author's assumptions, this shaft work is zero. Two frames of reference may be used by the observer and all quantities in the equation must be written as seen and felt by the observer. If the observer is at rest with respect to the tube wall, he feels only the earth's gravity, and the gravitational work is zero. If the observer rests in a frame of reference rotating with the fluid, he experiences the centrifugal field, and therefore the gravitational work will have a value, but the velocity seen by the observer will be reduced so that the sum of the kinetic energy and the gravitational energy will be a constant. The result of the energy equation, by any correct method, is to show that the sum of the enthalpy and the kinetic energy is a constant, if friction and heat transfer are omitted.

The method of the author gives an energy transfer of zero.

Newman Hall, University of Minnesota

Mr. Webster's analysis suggests one possible approach. In this treatment he wisely points out that his treatment cannot explain the full details of the flow and that it leaves several observed features inadequately explained. It would appear, then, that such a simplified theory will certainly need to be extended greatly to be adequate.

In reexamining this approach, it is suggested that several factors be considered:

1) Use of one-dimensional flow analysis in a field where two- and even three-dimensional behavior is occurring is very precarious. A much more careful consideration of the interaction of adjacent streamline and fluid elements than is given here is essential.

2) Action of pressure gradients and inertial force fields with respect to the motion of fluid elements requires extreme caution in analysis to avoid confusion as to the transfer of work within the fluid mass.

3) Identification of local temperatures or even stagnation temperatures with an observed cooling effect requires more careful specifications as to the precise nature of the correlation between analysis and experiment involved. This

is particularly true when rotational as well as translational motion occurs.

4) It is believed that the omission of any viscous effects may be one of the more serious weaknesses of the approach. At some point within the flow, it is very probable that velocity gradients, perhaps between the hot and cold streams, of sufficient magnitude occur to require consideration of viscosity.

5) The actual mechanism of separation of the hot and cold streams is almost entirely overlooked. This is certain to be a major item in a more detailed study.

M. M. Bolstad, University of Missouri

In my opinion, the point that there is constant angular velocity in the vortex is not justified. The effects of wall friction and momentum exchange radially probably result in the actual flow being somewhere between constant angular velocity and constant angular momentum.

Possibly the performance of the tube could be explained like this: the air entering the nozzle at ambient temperature and high pressure undergoes a big temperature drop as the pressure falls through the nozzle and its velocity increases greatly, the exact amount depending on the ratio of the outlet pressure to inlet pressure and the type of nozzle (converging or converging-diverging). The high velocity air continues to expand as it spirals inward around the main tube. If there were no wall friction, shock waves, or momentum interchange between "successive layers" of the spiral, the continued adiabatic expansion would result in further temperature and pressure drop and velocity increase until the air reached the center of the tube. For example, with air entering the nozzle at 70 F and 30 psig, it would attain a static temperature of —74 F in expanding isentropically down to atmospheric pressure.

Actually there certainly is wall friction, momentum exchange between "layers," possibly shock wave formation, and the flow is probably far from isentropic as implied in the paper. Due to the increasing velocity approaching the center, the momentum exchange results in each layer of gas doing work on the preceding layer and having work done on it by the next layer of gas closer to the center of rotation. This, then, along with the wall friction and possible shock, results in a final equilibrium where the temperature at the center is not as cold as if the expansion were isentropic. (There may be an additional temperature drop as the cold air expands through the orifice.) Furthermore, the work done by each "layer" on the preceding "layer" accounts for the temperature increase away from the center of the tube and the warm air going out the one tube.

C. M. Ashley, Carrier Corporation

The validity of this theory depends upon the demonstration in the first 12 equations that the *angular* velocity of the stream remains constant as the fluid spirals in toward the center of the jet— i.e., the linear velocity varies directly with the radius. Unfortunately, the validity of this broad conclusion, when unrelated to frictional losses within the stream, can be challenged and along with it the whole of the theory of which it is the cornerstone.

In the first place the conclusion is refuted by observed facts—when you pull the stopper in a wash basin, the vortex which is formed very evidently conforms more closely to the "free vortex" condition—that is, velocity inverse to the radius—than to the constant angular velocity condition postulated by the author. While the author maintains that the constant angular effect is due to the compressibility of the gas, the mathematics do not contain a density term. Furthermore, the vortex flow at velocities approaching the speed of sound in the vaneless diffusers of centrifugal compressors can be predicted with good accuracy using the free vortex theory but not by the constant angular velocity theory.

Another line of approach is an examination of the mathematics used by the author. Referring to Figure 2c and Equations 5 and 6, the author shows the radial work done *by* the gas element and the tangential work done *on* the gas element, whereas to be mathematically consistent he must show both components of the work done *by* the gas element. This can be understood clearly by pointing out that, since the sum of the two work components is in a direction at right angles to the motion of the element, the value of the sum must be zero. Because of this inversion of sign, a corresponding inversion of the power function of the velocity is made in Equation 7a which should show the velocity varying as the —1 power of the radius which is inverse instead of the +1 power, which is direct.

Another point which deserves mention is that the author's mathematics involve no dimensionality of the tube but we could certainly expect that the results would not be the same with a tube 0.1 in. long as with 10 ft nor for a tube 1 ft in diameter as compared to 1 in.

In an ideal free vortex, available energy of each element of the stream is transformed into kinetic energy with resulting decrease in the static temperature of the stream. However, when the stream is once more brought to rest, the temperature resumes its original value since the total energy has not been changed. To effect a change in total temperature, it is necessary to transfer energy from one portion of the stream to another but the author has demonstrated no mechanism by which such transfer of energy can be accomplished.

It can be shown that at the center of a jet the vortex velocity does approach a constant angular value but for a different reason than that proposed by the author—i.e., internal fluid shear forces.

Author's Closure

As judged by the foregoing comments, this paper was at least successful in its secondary aim of stimulating discussion of the vortex tube.

Professor Bailey and his coworkers appear to have carried out rather extensive researches on vortices, and the summary of their results to date is good, though necessarily brief. The experimental work, however, was carried out with a type of tube in which there was only one flowing stream, the air being introduced tangentially at the closed end and then flowing down to the open end. It is hoped that they will extend their work to the type of tube considered in this paper, that is, a two-stream tube in which the cooled stream leaves immediately at the closed end and the hot stream alone passes down the length of the tube. In reference to the statements made concerning Equation 14: Since the first part of the treatment was concerned with developing a recoil mechanism for radial and tangential decelerations, it would be expected that the use of this mechanism would result in the same work equation as for a centrifugal machine, and consequently that the agreement on pressures and temperatures is not pure coincidence.

In his first paragraph Professor Goetz questions the applicability of thermodynamics to this problem, on the premise that equilibrium is not attained among the molecules during their brief residence in the tube (10^{-3} to 10^{-4} sec). However, in view of the fact that a single molecule makes on the order of 10^{10} collisions per second under the temperatures and pressures in question, it would consequently undergo 10^{6} to 10^{7} collisions while in the tube. This is far in excess of the number of collisions required to establish equilibrium, and it is consequently likely that at any point equilibrium among adjacent molecules is closely approached. Also, as an empirical check, it can be noted that the laws of thermodynamics adequately describe the behavior of orifices and impact tubes so small that the action period is around 10^{-5} sec. As for viewing the behavior of the tube in terms of the kinetic theory, it certainly is permissible, but most likely would present mathematical difficulties, at least for the writer.

Mr. Fulton may very well have located the weak spot in my analysis, but at present I will have to take his conclusions on authority and hope that he gives a fuller exposition of his ideas in his forthcoming paper. Mr. Ashley's criticism follows the same general line of thought, so will not be dealt with separately. In the event that the approach to constant angular velocity measured by Professor Bailey is caused by internal friction rather than my proposed mechanism, still, the pressure and temperature relations from Equation 19 on should be useful as a means of correlating results on the tube.

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The Use of the Expansion of Gases in a Centrifugal Field as Cooling Process*

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The design of a vortex tube of good efficiency in which the expansion of a gas in a centrifugal field produces cold is described. The important variables in construction and operation are discussed and data for several tubes under various operating conditions are given. Low pressure gas, 2 to 11 atmospheres, enters the tube and two streams of air, one hot and the other cold, emerge at nearly atmospheric pressure. The cold stream may be as much as 68°C below inlet temperatures. Efficiencies and applications are discussed.

1. INTRODUCTION

The adiabatic expansion of a gas with external work, which may take place in a reciprocal engine or in a turbine, is the most effective cooling process. It requires, however, a considerable technical effort, especially in the region of low temperatures, where many technical difficulties arise. In the technique of gas liquification, therefore, the expansion with external work is usually replaced by the use of the much simpler throttle effect, which does not require any moving parts. This cooling process is much less efficient, and is only possible because real gases deviate from the perfect gas law.

Under these circumstances it seems important to investigate an arrangement which was described by Ranque in 1939.[1] In this arrangement compressed air is permitted to enter through a tangential nozzle into a cylindrical container, formed by a tube of 12 mm diameter. A turbulent flow of gas, in a screw-like motion, escapes through both ends. This rotating air stream produces a region of increased pressure near the wall inside the cylinder, and a region of decreased pressure near the axis. If one end of the cylinder is closed by a diaphragm which permits the escape of air only from the central region, while the other end is throttled, the air escaping through the central diaphragm has a reduced temperature, while the air escaping through the other end shows a temperature increase. The observed temperature difference between the two streams, using compressed air of 6-atmos. gauge pressure at 20°C, was 70°C, while the lowest temperature of the cold stream was −12°C.

The purpose of this investigation is to design an arrangement of this kind which can be used as a cooling process of maximum efficiency, and to determine the largest efficiency obtainable.

2. CONSTRUCTION AND FUNCTION OF A "VORTEX TUBE"[2]

The results of numerous preliminary experiments showed that for a favorable cooling effect the diaphragm should be installed as near as possible to the nozzle. The circular air flow near the nozzle and the diaphragm should be as nearly as possible of rotational symmetry. This is achieved by a peculiar method of introducing the compressed air, which is shown in Fig. 1b and 1c. Two pieces of tubing of internal diameter R are attached to the cylinder by means of flanges and overlapping nuts; one of the flanges acting as diaphragm B located near the nozzle.

For an explanation of the mechanism of the vortex tube we refer to Fig. 1a. If both ends of the attached tubes are open, the air entering tangentially through D escapes with a screw-like motion along the wall of the left tube, while the centrifugal force and the internal friction of the gas produce a lower pressure in the axial region. Hence, air will be sucked in through B from the right tube. This can be avoided if the flow through the left tube is throttled by a valve attached to the end of this tube. This valve should be sufficiently far away from the nozzle D (about 50 R) so that the gas reaching it will have lost most of its screw-like motion because of internal friction. By partial closing of the valve, it is possible to force a fraction, μ, of the air stream to escape to the right through the diaphragm B. This fraction increases with increasing internal pressure p_i which is measured in the left tube near the valve, and it originates in the region near the axis of the vortex in the left part of the tube. The air escaping through B has been expanded in the centrifugal field from a region of high pressure near the wall of the cylinder to a pressure p_i in the region near the axis; and it has transferred a considerable part of its kinetic energy by means of internal friction to the peripheral layers. These flow to the right and acquire a higher temperature. Hence, a fraction μ of the expanded air leaves the tube to the right with reduced temperature, and a fraction $(1-\mu)$ escapes to the left with increased temperature.

In the absence of internal friction, and with a sufficient pressure gradient, the velocity of the air would increase to supersonic speed during the expansion between the circumference and the axis of the vortex tube. The internal friction, however, is particularly effective in this region. It causes a flow of energy from the axis to the

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* Submitted by: R. M. Milton, Department of Chemistry, Johns Hopkins University; Translated by: I. Estermann, Physics Department, Carnegie Institute of Technology.

[1] M. G. Ranque, J. de. phys. et rad. [7], 4, 112 (1933).

[2] Literal translation of the German word "Wirbelröhre."

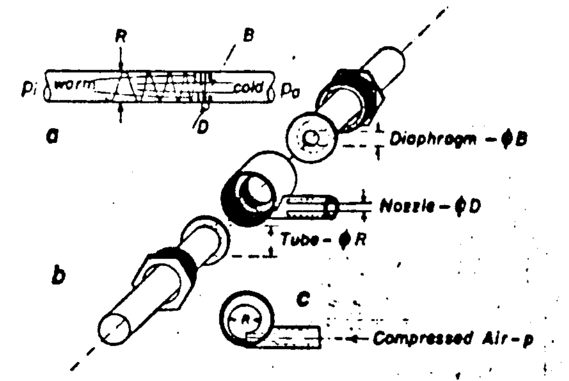

FIG. 1. Vortex tube; (a) schematic sketch; (b) tube taken apart (thread surfaces black); (c) central piece with nozzle.

circumference by trying to establish a constant angular velocity throughout the cross section of the tube. There will be, therefore, a decrease in the heat content of the axial part of the air stream flowing to the right and an increase in the heat content of the stream flowing to the left, if heat exchange with the surroundings through the wall of the tube is prevented.

The demonstration of such a vortex tube is simply astonishing. With proper choice of the fraction μ and $1-\mu$, compressed air of a few atmospheres pressure and 20°C will easily produce a temperature of +200°C in the left tube and −50°C in the right tube.

3. DETERMINATION OF THE MOST FAVORABLE CONDITIONS FOR THE OPERATION OF A VORTEX TUBE

In the construction of a vortex tube according to Fig. 1 the results of several preliminary experiments have been utilized. It appears that the efficiency is not affected by any particular shape of the cylindrical container, or of the "warm" tube. We used, therefore, a smooth cylindrical tube of sufficient length. The tube carrying the diaphragm B may have any shape which does not interfere with the flow of the cold air. Attempts to increase the efficiency by using diaphragms of special shapes had no apparent results. The use of a thin-walled German silver tube and the arrangement of screws and flanges for the assembly as described earlier reduces the heat transfer between the warm and the cold part to a small value.

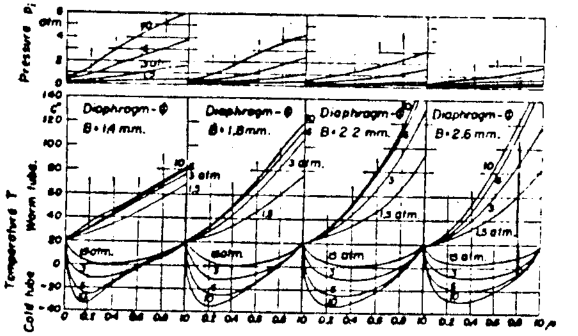

FIG. 2. Temperature of cold and hot gas stream and internal pressure in tube No. 1 as function of external pressure, size of diaphragm, and μ. μ = mass of cold/mass of total air. Constants: $R = 4.6$ mm, $D = 1.1$ mm, $T_0 = 20$°C, $p_o = 1$ atmos. Parameter: pressure p at the nozzle.

While the general design is fixed by the simple principles listed above, there are a number of variables to consider which affect the operation. Since it seems rather difficult to derive these influences theoretically from an analysis of the behavior of the compressible flow in a quantitative form, they were subjected to an experimental investigation. The temperatures of the gas in the two flow directions depend on the following variables:

(1) Temperature, T_e, pressure p; and rate of flow G of the expanding compressed air.
(2) External pressure p_o outside the vortex tube.
(3) Diameters of the tube (R), the diaphragm (B), and the nozzle (D); these quantities determine G.
(4) Ratio of distribution of air between the flow in both directions, μ-mass of cold air/mass of total air.

The measurement of the temperature of the streaming gas streams presents a separate problem. Thermocouples arranged in the gas stream would give incorrect results, since it is certain that the expanded cold air has not completely given off its kinetic energy and still contains a rotating flow. For this reason, the temperatures T were simply measured at the thermally insulated walls of the tubes. These measurements yield temperature differences which are certainly not too favorable, but will probably be too small.

4. RESULTS

Figure 2 contains results obtained with a vortex tube of a diameter $R = 4.6$ mm. The effect of the nozzle diameter D cannot be ascertained from this figure, but it was found by experiments that $D = 1.1$ mm was a favorable value. Smaller values of D would certainly be less favorable. If they were chosen, the rotating flow would decrease too fast on account of the small rate of flow and the more important contribution of the friction between gas and walls. Larger values of D would result in such large rates of flow that it could not be carried by the tube without increasing the pressure p_i to too large values. This would make the expansion ratio p/p_i unfavorable. The pressure difference $p_i - p_o$, moreover, causes losses at the throttle.

Figure 2 shows the temperatures of the cold and warm tubes as functions of μ for four different values of the initial gas pressure p. The ratio μ can be changed by means of a valve attached to the free end of the warm tube, which was about 30 cm long. At such a distance from the nozzle, the rotating component of the gas motion had decreased sufficiently. The value of μ was measured with two gas meters arranged in the cold and warm stream, respectively. The total rate of flow is proportional to the absolute pressure p of the entering gas, but is practically independent of the setting of the valve and, therefore, of μ. For $D = 1.1$ mm and air, the rate of flow was, for instance, $G = 7.0$ m³/hr. for $p = 10$-atmos. gauge pressure. The internal pressure p_i was measured near the throttle valve and is also plotted in Fig. 2. The influence of the diameter of the diaphragm B is shown for four different values.

There is always a certain value of μ producing a minimum value of the temperature of the cold air. The larger the value of μ, the higher the temperature of the warm end. A small diaphragm ($B = 1.4$ mm) is rather unfavorable, especially at high values of μ, since then the internal pressure p_i and the throttle losses become too large. On the other hand, the largest diaphragm ($B = 2.7$ mm) gives a smaller cooling effect in the minimum of the curve because it permits the entrance of air from regions where the rotating flow is still considerable. For large values of μ (e.g., 0.8), the most favorable cooling effect is obtained with the largest diaphragm. The choice of the diaphragm diameter B depends, consequently, on whether it is desired to reach very low temperatures or to produce large quantities of cold air. The value $B = 2.2$ mm may be considered as a most favorable compromise solution for these two factors.

Many more details may be seen in the curves. It should be pointed out, in particular, that the cooling of the cold air and the heating of the warm air with respect to room temperature are equal for $\mu = 0.5$. It is easy to see, moreover, that the relation

$$\mu \text{ (cooling)} = (1-\mu) \text{ (heating)}$$

must be always valid since the amount of heat removed from the cooled air must be equal to the amount of heat given to the heated air. It should also be remarked that the setting of the throttle valve giving minimum temperature can be found easily without special measurements; a boiling sound is audible if the valve is set for the correct value μ (e.g., $\mu = 0.3$ for 10 atmos. and $B = 2.2$ mm, $T = -35°\mathrm{C}$). It should be realized that the temperature of the cold air is measured too high if μ is very small, and the temperature of the warm air is measured too low if μ is near 1, because in both cases the rate of flow through the tube in question is very small.

It would be necessary to repeat all the measurements for different values of the tube diameter R, in order to investigate the influence of this quantity. It is, however, sufficient to report the measurements in the cold air stream with the most favorable value of the diameter of the diaphragm B for a wider tube (Fig. 3). The diameter of the tube R has been increased to more than twice its former value, and the nozzle diameter D in proportion. In this case, the temperatures of the cold air are lower than those in Fig. 2 for all values of p. Additional data for the lowest temperatures obtained with a still larger tube as function of p are given in Fig. 4. The total rate of flow G for all tubes may be represented by the relation

$$G = \text{const.} \cdot F \cdot p,$$

where F is the area of the nozzle in mm^2, p the initial air pressure in atmos., G the rate of air flow in kg/hr., and const. $= 0.80$ kg/(h mm^2 atmos.).

Finally, we show in Fig. 3 the product $\mu(T_0 - T)$ as function of μ for tube No. 2. This product is proportional to the total cooling effect. It is apparent that the maximum heat removal does not occur at the value of μ for which the lowest temperature is obtained.

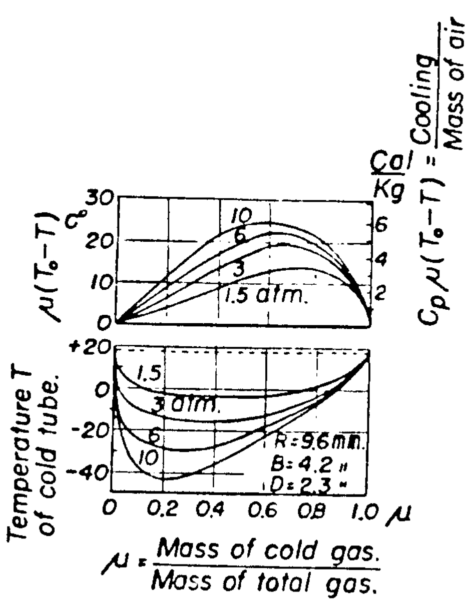

Fig. 3. Temperature and heat content of the cold air stream of tube No. 2 with most favorable diaphragm.

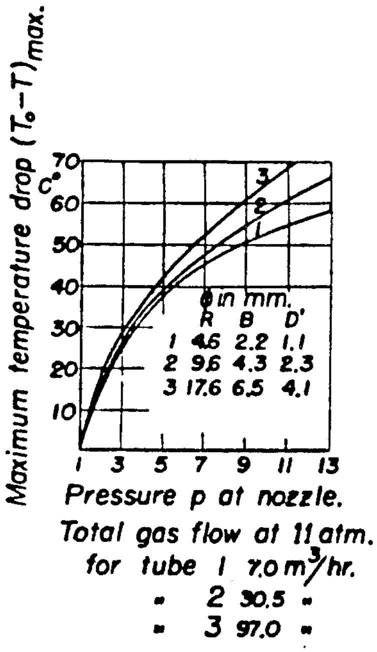

Fig. 4. Maximum temperature drop for tubes Nos. 1, 2, and 3.

It follows from Fig. 4 that increasingly larger temperature reductions and cooling effects may be obtained with larger vortex tubes at the same initial pressure. The rates of flow, however, become considerable, and the capacity of the compressor which was available did not permit the use of larger tubes. It is easy to see that the efficiency increases for larger tube diameters, because for larger tube diameters the heat losses due to heat conduction from the axis to the circumference of the vortex tube become less important. The larger the tube diameter R, the smaller is the ratio B/R, where B is the most favorable diameter of the diaphragm. It should be remembered, however, that a relatively small diaphragm favors transmission of cold air which has been well expended and is practically free from revolving motion.

5. THERMODYNAMIC EFFICIENCY OF A VORTEX TUBE

The calculation of the cooling efficiency of a vortex tube will be carried out by comparing it with the cooling efficiency of the adiabatic expansion of a perfect gas in an expansion engine delivering external work. Such an engine will, theoretically, deliver a continuous flow of cold air of the temperature T_1, produced by expanding a gas of the temperature T_0 from the pressure p to the pressure p_a. For this process,

$$T_0/T_1=(p/p_a)^{(\gamma-1)/\gamma}. \tag{1}$$

The work E delivered by the expansion engine per unit mass is $E=c_p(T_0-T_1)$ and is equivalent to the amount of heat Q removed from the gas. The minimum amount of work required for (isothermal) the compression is

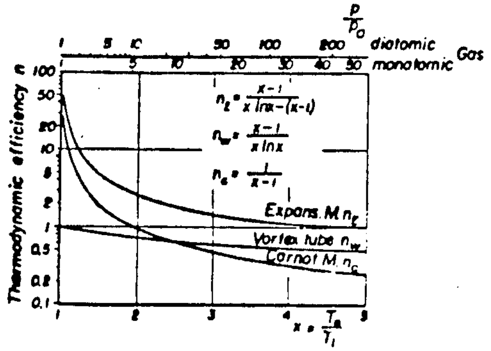

FIG. 5. Efficiency of 3 refrigerating machines for comparison.

$$A=RT_0\ln\frac{p}{p_a}. \tag{2}$$

Substituting for p/p_a from Eq. (1), one obtains

$$A=c_pT_0\ln\frac{T_0}{T_1}. \tag{3}$$

The net work required is $A-E$, and the result obtained for the efficiency of the expansion engine is

$$\eta_E=\frac{Q}{A-E}=\frac{T_0-T_1}{T_0\ln(T_0/T_1)-(T_0-T_1)}=\frac{x-1}{x\ln x-(x-1)}, \tag{4}$$

where $x=T_0/T_1$.

The efficiency of the vortex tube is less than that of the expansion engine, since only a fraction μ of the gas is expanded into a cold stream and heat is removed by another fraction of the gas $(1-\mu)$. Moreover, no expansion work is gained.

According to the explanation of the mechanism given in Section 2, the gas is expanded along a spiral path and withdrawn through a diaphragm. The internal friction tends to produce a uniform angular velocity along the path of expansion in the centrifugal field. The same effect could be obtained by installing a turbine wheel in front of the nozzle. This would transform the vortex tube into an expansion turbine, where the compressed gas enters near the circumference and leaves at low pressure near the axis while mechanical work is delivered by the turbine rotor. In this theoretical case all the gas could be removed as cold gas through the diaphragm; and the efficiency would be η_E as derived in Eq. (4). The work developed at the turbine rotor could be changed into heat by a friction clutch on the shaft. This heat could be removed by branching off a very small fraction of the expanded gas towards the warm side of the vortex tube which would escape with any desired high temperature.

The efficiency of such a hypothetical vortex tube with a hidden turbine would be larger than that of a real vortex tube, since almost all the gas would escape as a cold stream through the diaphragm. No expansion work would be developed in this case; its heat equivalent would be carried off by the minute amount of gas escaping through the warm tube. The efficiency of such an ideal vortex tube shall be compared with that of the actual vortex tube; it is obtained by omitting the expansion work E in the denominator of Eq. (4). The result is

$$\eta_W=(x-1)/x\ln x. \tag{5}$$

Figure 5 shows the dependence of η_E and η_W on x, also, and for comparison only, the efficiency of a Carnot engine η_c. While such an engine is not very practical for the production of a continuous stream of cold air of the temperature T_1, it is frequently used in practice for such a purpose. The upper scales in Fig. 5 show the pressure ratio p/p_a for which the temperature ratio T_0/T_1 is reached for mono- and diatomic gases. For $x=1$, η_c and η_E approach ∞, while all values of η approach zero for large values of x.

The theoretical efficiency of the vortex tube is rather favorable, expecially for large pressure ratios where it approaches the value η_E.

The results of these theoretical considerations may now be compared with data for actual vortex tubes found by experiment (Table I). The data given in this table refer to the largest tube No. 3 with a diameter $R=17.6$ mm. Data given in line 2 are taken from the curves in Fig. 4. The values for the quantity $[\mu(T_0-T)]_{\max}$ in line 5, which is proportional to the amount of cold Q, are taken from the curves in Fig. 3 and have been multiplied with a factor 1.1 (see Fig. 4), in order to obtain corresponding values for

TABLE I. Efficiency of tube No. 3. $R=17.6$ mm, $T_0=18$°C, $p_a=1$ atmos.

1. Pressure ratio used (p/p_a)	2.5	4	7	11
2. Maximum temperature drop (T_0-T)max	24°C	36°C	52°C	68°C
3. Theoretical temperature drop (T_0-T_1)	67°C	95°C	124°C	144°C
4. Temperature efficiency $\alpha=(T_0-T)\text{max}/(T_0-T_1)$	0.36	0.38	0.42	0.47
5. $[\mu(T_0-T)]$max	15.5°C	21°C	24°C	27.5°C
6. Cooling efficiency $B=[\mu(T_0-T)]\text{max}/(T_0-T_1)$	0.23	0.22	0.19	0.19
7. Theoretical efficiency η_W	0.88	0.81	0.77	0.72
8. Practical efficiency $B\eta_W$	0.20	0.17	0.15	0.14

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tube No. 3. It is apparent that the obtained temperature drop α reaches almost 50 percent of the theoretical value. The "cooling efficiency" B is defined as the ratio between the actually obtained efficiency and the theoretically calculated efficiency, it amounts to about 20 percent. The practical thermodynamic efficiency in line 8 is defined as B_{NW}; this definition would be valid if an ideal air compressor would be used.

It is probable that the real efficiencies are somewhat better than the data in Table I indicate. This is caused by, firstly the method of temperature measurement which gives unfavorable values, and secondly, to the presence of moisture in the compressed air, which appears in the cold stream as a fog of ice crystals.

6. APPLICATIONS OF THE VORTEX TUBE

Since results for the efficiency of a well-designed vortex tube operated with compressed air are now available, it may be in order to discuss possible applications.

There is little probability that vortex tubes will replace the customary refrigerating machines, since their efficiencies are much better in the region of small pressure ratios. There may, however, be special cases where a vortex tube would be more desirable because of its simple construction (e.g., air cooling in mine shafts).

The situation is different at low initial temperatures and high pressure ratios. It may be expected that the vortex tube will be superior to the throttle expansion (Joule-Thompson effect) for gas liquefaction. For these applications to lowest temperatures additional measurements are planned and will be reported later.

COMMENTS ON THE VORTEX TUBE

C. D. Fulton
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ED. NOTE: *This discussion, although written to apply specifically, to the theory advanced in the article on the facing page, brings up several points of general interest concerning all research on the Vortex tube. For this reason, and because Mr. Fulton writes with such enthusiasm about the "Ranque Society," the discussion is published as an entity within itself rather than as a final comment tacked to the end of the article, the customary publication procedure.*

The revival of Ranque's tube begun by Hilsch in 1945 brought it to light for the first time. From 1946 to 1948 there was talk of revolutionizing the refrigerating industry. Then came the awareness of the tube's extremely low efficiency and of the unlikelihood that it would ever become useful. That, one might have thought, would have sent it back into limbo. On the contrary, more, not fewer, students are studying it in the schools; more industrial laboratories are investigating it; more publications are appearing about it; and it has reached the state of being the object of sponsored research.

Ranque's tube, the problem child of the mid-century, seems to be here to stay. It is time to give a name to the growing army who baby-sit for it instead of going to the movies. While "suckers" is possibly the right name for us, we would probably prefer to be called the Ranque Society—on an informal basis, of course, with no dues. No more catholic group could be found in the technical world for we include every brand of engineer, and physicists, chemists, metallurgists, and every permutation thereof. How long this will go on, and where it will lead, if anywhere, no experienced member of the Society would any longer dare to say. Ranque's tube commands a fascination—a hope, a curiosity, a scientific interest, a childish joy—that hangs on and on. Perhaps some day it will be said that never have so many made so much noise about so little. Perhaps on the other hand some day the Society will be decorated by the government.

At least 10 theories have been proposed during the last six years on the operation of the Vortex tube. Some of them turn out to say nothing at all, throwing the burden of the argument on an undefined phrase like "centrifugal expansion," "molecular sifting," or the like. In some others, equations have been tortured to show two different temperatures, while in still others the mass of manipulations has grown so great as to be incomprehensible. It is certainly true to say that "scientists are disagreed as to what makes the tube work."

In this state of confusion which, if anything, is increasing, we need now to fall back to where we should have started in the beginning. A traverse of the pressures, temperatures, and velocities inside the tube is certainly the most down-to-earth starting point for an understanding of how it works. This is in accord with good scientific procedure. Mr. Scheper therefore deserves credit for driving an opening wedge into this mystery. It is too bad that the usual tubes are so small as to render these measurements very difficult. The experimenter is often forced into building ungainly tubes which he usually has insufficient compressed air to operate, or which have too small a nozzle for the other dimensions. It looks as if the whole story of the internal flow will only be obtained by (1) using probes, etc., of tiny size, a watchmaker's job, or (2) operating a large correctly designed tube from a huge air supply such as the sort of compressor used in a gas turbine.

Mr. Scheper's results surprise me in two respects. First (Fig. 1) he finds an outward flow from the core approaching the orifice, except right at the orifice plate. This is at variance with my previous flow pattern, shown in the May 1950 issue of REFRIGERATING ENGINEERING. Second (Fig. 3) he finds that, near the nozzle, the static temperature has a minimum about ¾ of the way from the axis to the tube wall.

Mr. Scheper's theory seems to say something and to say it well enough to make a showing. As to two distinct streams rubbing upon each other over an area, this seems a reasonable concept, but the situation is somewhat clouded by the recirculations and the doubling back of the core flow. What is needed to clarify this matter is a thorough traverse of the velocity field so that a detailed flow pattern can be drawn. In fact, there is nothing that would be more useful, at the present time, than an authentic flow pattern.

The main bone of contention in Mr. Scheper's theory will be the temperature difference. The questions, as I see them, are as follows:

1) Is there an appreciable static temperature difference in the right direction?
2) How does this temperature difference arise?
3) Does this temperature difference act over a large enough area to transfer the heat?
4) What does a static temperature gradient mean, anyway, when there is also a large velocity gradient? Is the heat transfer proportional to the gradient of static temperature, or to the gradient of something else such as stagnation temperature or adiabatic wall temperature?

As to the first question, the lowest curve in Figure 3 ought to show this static temperature difference if it exists. But that curve shows a uniform core temperature of 43 F up to about one-half the tube radius, with a minimum of 39 F beyond that. Thus a difference of 4 degrees is all that is at work, and to say even this requires the choice of the minimum temperature of the outer layers, and not their average temperature. Farther down the tube, at axial station 2, can one construe the temperature gradient as helpful at all? The measurements, therefore, do not seem to bear out the large static temperature difference Mr. Scheper assumes in constructing his theory. He seems to assume that the outer layers have the low static temperature resulting from an isentropic nozzle expansion, while the returning core flow has a higher temperature because it has been decelerated with constant energy, more or less. (Mr. Scheper's assumptions here are not clear. They need discussion, under question 2 above.) There is, in summary, a grave conflict between the measured and the assumed static temperatures.

As to the third question, whatever favorable static temperature exists does not persist far down the tube, according to the measurements. Therefore it appears that much too large an area has been ascribed to the heat flow.

Wi[illegible] alterations, would not the calculated film coeffi[illegible] enormous, and therefore unbelievable?

The fourth question is profound, but it ought to be attacked.

Before the appearance of Mr. Scheper's theory, there were, in my opinion, two respectable theories in the field:

The theory of the centrifugal flow of kinetic energy because of friction acting on the free-vortex velocity distribution (Ranque, Hilsch, Fulton).

The theory of the conversion of the free vortex into a forced vortex (Kassner & Knoernschild).

If Mr. Scheper's theory is destined to join this list, it will need able defense and continued development. Whatever the outcome, he has given us something to think about. And his measurements of conditions inside the tube are a welcome opening move in the one and only reliable method of solving the problem of Ranque's tube.

THE VORTEX TUBE—Internal Flow Data and A Heat Transfer Theory

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General Electric Company

This article continues the discussion, begun two years ago in REFRIGERATING ENGINEERING, of the thermodynamic principles which cause the hot and cold air flows through what has become known as the Hilsch Vortex Tube. The earlier articles are:

"Analysis of the Hilsch Vortex Tube" by D. S. Webster, published in February 1950

"Ranque's Tube" by C. D. Fulton, published in May 1950

"Fluid Action in the Vortex Tube" by Roy MacGee Jr., published in October 1950

"Bibliography of the Vortex Tube" by Walter Curley and Roy MacGee Jr., published in February 1951

THE vort[illegible], or Hilsch, or Ranque, tube as it is variously [illegible]ed is a remarkably simple device which produ[illegible] hot and cold gas streams simultaneously from a so[illegible] of compressed gas. This device has no moving pa[illegible] It consists of a straight length of tubing with a conc[illegible]ic orifice near one end and a nozzle entering tangentially near the outer radius adjacent to the orifice plate (see Fig. 1). By throttling on the end of the tube farthest from the orifice, a cold stream is forced through the orifice, while a hot stream issues from the opposite end of the tube. Cold air temperatures 100 deg F below nozzle temperature are readily obtainable at moderate nozzle pressures with correspondingly high hot air temperatures.

Interest in the vortex tube has been increasing rapidly in this country since the publication here of the experimental work done in Germany by Rudolf Hilsch[1]. In this basic work Hilsch determined tube proportions for optimum performance, and overall performance data over a wide range of nozzle pressures. The device was actually invented in France by G. Ranque, who obtained a French patent in 1932 and a U. S. patent in 1934. Ranque described the tube in an article in 1933[2].

That the vortex tube does actually separate a compressed gas into higher and lower stagnation temperature streams is beyond question. How it does this is at present a subject of much study and controversy. To the writer's knowledge no data have previously been published on the internal flow processes in a vortex tube.

As the subject of a master's degree thesis, the writer performed experiments on a vortex tube to define, if only approximately, the flow and temperature patterns existing under various operating conditions. This work formed the basis for the "heat transfer theory" advanced in the thesis, completed in May 1949[3]. The present paper incorporates much of the thesis work and presents the heat transfer theory in slightly modified form. This is, to the writer's knowledge, the first vortex tube theory based on forced convection heat transfer to be published.

On this theory, heat transfer occurs radially from the vortex core outward by virtue of static temperature gradients in that direction. The heat sink is provided by the outer gas layers which are at lower static temperature due to the nozzle expansion. This heat transfer raises the stagnation temperature of the outer gas which produces the hot flow, while at the same time the stagnation temperature of the core is lowered producing a cold flow through the orifice.

Test Results

Two classes of tests were used to determine the flow patterns. The first was by visual methods; the second by instrument traversing. In all of these tests a relatively large tube diameter (1½ in. ID) was used to facilitate probing the interior of the tube. The nozzle design and tube proportions were dictated by simplicity of construction and compressed air supply limitations. As was to be expected, these design compromises resulted in poorer overall performance than the tubes of Hilsch[1]. However, for the purpose of this investigation, optimum performance was not considered to be of importance.

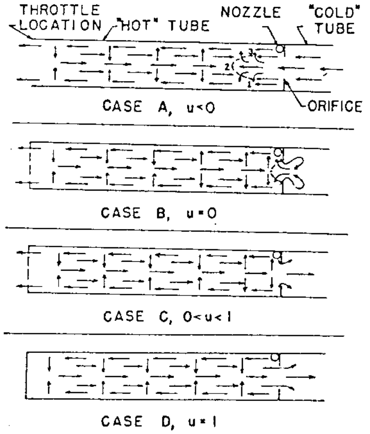

Fig. 1. Basic flow patterns for four operating conditions defined by cold fraction *u*.

For the visual testing the tube used was glass with a brass nozzle section and clamp ring. The "hot" tube was three feet long, and the "cold" tube was eight in. long. The nozzle diameter was ¼ in., and the cold orifice diameter was ½ in. The most effective way found to visualize the flow was to use silk thread knotted loosely around a tightly stretched wire running axially through the tubes. The thread was about ⅛ in. long, pointing radially, and free to rotate and slide axially along the wire. In operation, the thread rotated rapidly at all axial positions as long as the center of rotation (the wire) was within about ¼ in. of the tube axis. Outside of this diameter the thread did not rotate, but assumed a tangential direction.

The most revealing feature of these tests, however, was the axial motion of the thread at various radii and degrees of hot end throttling. These results are shown in Figure 1 by arrows parallel to the tube axis. In this figure, and as used throughout this paper, the symbol u represents the fraction of nozzle flow passing through the orifice, and its value is determined by the amount of throttling on the end of the hot tube. It is seen in Figure 1 that there exists for all values of u a definite and strong reverse flow in the core with a diameter approximately equal to the orifice diameter, and that this flow originates at a point very near the throttle end of the tube.

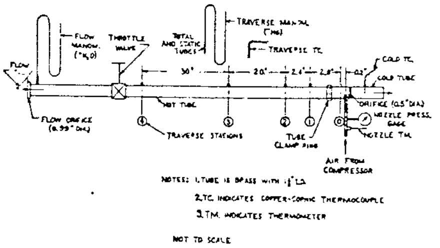

Fig. 2. Test setup showing instrumentation.

Chemical smoke testing with stannic tetrachloride was in general unsatisfactory due to the high degree of turbulence, even at the lowest nozzle pressures. However, the smoke did give some indication of the presence of a more or less solid rotation core flow.

Oil in the compressed air left a helical oil trace of increasing pitch toward the hot end on the tube wall. The angles of this helix indicate that near the tube wall the axial component of velocity is less than the tangential component for virtually the whole tube length.

From all the foregoing, and in particular the axial motions of the threads, a great deal can be deduced about the radial flows which occur in the tube. Figure 1 shows the four possible operating conditions of a vortex tube at constant nozzle pressure, varying only the cold fraction, u, by throttling the hot end. This figure summarizes the internal flow configurations in a qualitative form and is basic to the theory presented later.

In Case A ($u < 0$) a suction is created at the orifice due to insufficient throttling. Thus, room air is drawn into the tube. This air penetrates axially to a stagnation surface of revolution such as 1-2-3, shown by the dotted curve, across which there is no axial flow. Extending from the extreme hot tube end up to the surface 1-2-3 is a core back flow caused by a sink flow near the tube end. This back flow is then returned to the outer helix flow by a source flow as surface 1-2-3 is approached. Raising the nozzle pressure moves surface 1-2-3 toward the hot end, due to the increased suction effect. Throttling the hot end moves surface 1-2-3 toward the orifice.

In Case B ($u = 0$) the throttling is such as to move the surface 1-2-3 partially through the orifice so that no net flow exists through the orifice. There probably exists a small recirculation through the orifice with inflow near the center and on equal outflow leaking around the edge of the orifice. The back flow and recirculation within the tube remains essentially as in Case A.

In Case C ($0<u<1$), the only operating condition at which an outflow of cold air is produced, the throttling is increased to push a net flow through the orifice. The back flow maintains itself, but as u is raised the source flow near the nozzle decreases, probably due to the increase in the static pressure gradient near the nozzle, and the higher axial velocity of the core flow. It is probable that a leakage exists, as in Case B, around the edge of the orifice due to friction on the orifice plate surface lowering the velocity and hence the centrifugal force on this air, which allows the pressure gradient to force it inward.

In Case D ($u = 1$) the hot tube is closed off, forcing the total flow through the orifice. As in all the previous cases the core back flow is still present, but there is probably no longer a source flow near the nozzle—i.e., all radial flow is inward. The flow pattern is in this case very similar to that obtained for a cyclone separator[4] which is essentially the equivalent of a vortex tube with no hot end flow.

The flow in a vortex tube may be summarized as consisting of an outer helix flowing away from the nozzle, and an inner or core helix flowing back toward the nozzle end. The radial flow distribution depends on the amount of throttling on the hot end.

In the instrument traversing tests the glass tubes used in the visual tests were replaced by brass pipe of the same diameter, having traverse holes located along the pipe. Figure 2 is a sketch of this test setup and shows the location of the five traverse stations (numbered 0 to 4). All other instrumentation is also shown. Traverse readings were made at each of seven radial positions (numbered 0 to 6) spaced ⅛ in. apart, with position 0 at the axis of the tube. Three traverse instruments were used: thermocouple "tube," static pressure tube, total* pressure tube. Each of these tubes was made with a short length of 3/32-in. OD stainless steel tubing. For the thermocouple tube, an exposed fused junction of copper-copnic wire extended about ⅛ in. from the end of the tube. The impact tube was made with a No. 80 drill hole (0.0135 in. diam) normal to the tube axis and located 3/16 in. from the plugged end of the tube. The tube was provided with a direction arm to indicate the approximate direction of the impact hole. The static tube was made with a No. 80 drill hole located in the end surface of the tube, concentric with the tube axis. Before drilling the hole, the end surface was machined flat and perpendicular to the axis after plugging the end of the tube with brazing material. The principle of this static tube is that the pressure hole is located in a flat surface which lies parallel to the rotational flow in the vortex. This can of course be valid only if the radial velocity component is negligibly small, which the writer believes to be true in the vortex tube.

It is recognized that this instrumentation, particularly the static pressure tube, is rather crude and places certain limitations on the accuracy of the data herein presented. It is felt, nevertheless, that the results obtained are sufficiently accurate to show the basic internal flow pattern. The instrumentation used was chosen for its constructional simplicity. The design and construction of a more adequate instrumentation for this complex flow field was beyond the scope of the work undertaken. In this paper a representative part only of the original data obtained is presented. Reference 3 contains the complete data obtained.

Static and total pressure vs radial position was measured at four traverse stations along the tube. The total pressures were maximum values obtained by rotating the impact tube. Angles at which this reading maximized (stream angles) were not measured but it was observed that the stream angles were essentially tangential, indicating relatively small axial velocities. This result is in agreement with the helical oil traces observed in the visual tests discussed previously.

Velocities were calculated on a compressible flow basis with the aid of Reference 5. The thermocouple temperatures were corrected for impact recovery by the use of a recovery factor of 0.65[6].

At axial stations 1 and 2 the flow field is essentially one of solid rotation up to about ⅛ in. of the tube wall (radial position 5). Between this radius and the wall the velocity distribution approaches that of a free vortex in which tangential velocity is inversely proportional to radius. The velocities obtained are essentially equal to the tangential components. At the axis of the tube, however, the axial velocity component dominates and hence the measured velocity does not go to zero. There may also be some lack of rotational symmetry.

Figure 3 is a plot of total and static temperature vs radius for a nozzle pressure of 15 psig and cold fraction (u) of 0.26. At this condition the cold air temperature was 43 F with a nozzle air temperature of 64 F. The significance of these curves with respect to the heat transfer theory of operation lies in the fact that the cold air stream which emerges from the core of the tube has a higher static temperature than the surrounding outer helix.

Figure 4 is a plot of total temperature vs axial distance from the nozzle for radial positions 1 and 5 which were chosen to be representative of the core flow and outer helix flow respectively. The arrows on the curves indicate the axial flow direction at these two radii. In this case cold fraction (u) was zero, the nozzle conditions being identical to those of Figure 3. It is observed from Figure 4 that cold temperatures are produced even though no flow goes through the orifice. In accordance with the requirement of an overall heat balance, assuming no heat transfer between the air and the metal of the tube wall, there is no net increase in temperature of the outer helix flow which issues from the throttle valve. Figures 3 and 4 lead immediately to the heat transfer theory of the vortex tube.

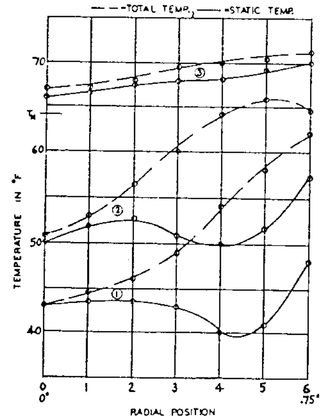

Fig. 3. Static and total temperature vs radius at three traverse stations.

Before leaving this section on test results it should be pointed out that the remainder of the data for pressure, temperature, and velocity obtained in the original research at some other operating conditions is very similar to the curves herein, the main difference being a general shift of the magnitudes at a given traverse station.

Theory of Operation

It has been shown that the flow pattern in a vortex tube consists basically of two counter current axial flows, with an approximately solid rotation. The inner core axial flow emerges from the orifice with a reduced total or stagnation temperature. The axial flow of the outer helix emerges from the opposite end of the tube at the throttle valve with an increased total temperature. The core is at a higher static temperature than the outer helix. These

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This paper presented before the Heat Transfer and Fluid Mechanics Institute, Stanford University, June 20-22, 1951.

This paper is based on a thesis for a master's degree, written by the author while attending Union College, Schenectady, N. Y., and is not related to any work of the General Electric Company.

* Total, stagnation, and impact are synonymous terms as applied to pressure or temperature.

conditions form the basis for the heat transfer theory of the vortex tube, which follows.

Referring to Figures 3 and 4, the increase of the total temperature of the outer flow is attributable only to heat transfer from the inner core, since no mechanical work is involved. The core, being at a higher static temperature than the outer helix, transfers heat by forced convection to the outer helix, and as a consequence reduces its own total temperature. The analogy to a concentric pipe counterflow heat exchanger is evident. If we imagine a zero thickness tube wall, having a diameter approximately equal to that of the cold orifice, which is rotating at every axial position at the same angular velocity as the vertex, the heat transfer across this imaginary tube surface is proportional to the bulk static temperature difference and to the overall film coefficient of heat transfer occasioned by the relative axial velocity of the two oppositely moving streams. With the assumption of such a rotating tube, the tangential velocities do not contribute directly to the film coefficient of heat transfer, nor does the flow field become altered, as would be the case with a stationary tube due to tangential wall friction.

NOMENCLATURE

A = effective surface area, sq ft
C = specific heat at constant pressure, Btu per lb, deg F
N = cooling effectiveness, dimensionless
P = nozzle inlet pressure, atmospheres absolute
Q = heat flow rate, Btu per hr
T = total (stagnation) temperature, F
t = static temperature, F
u = cold flow fraction, dimensionless
U = overall heat transfer coefficient, Btu per hr, sq ft, deg F
w = weight flow rate, lb per hr
θ = heat transfer mean static temperature difference, deg F
Δt = isentropic nozzle temperature drop, deg F

Subscripts on T or t

N = at nozzle discharge
C = cold flow at orifice discharge
H = hot flow at throttle valve discharge

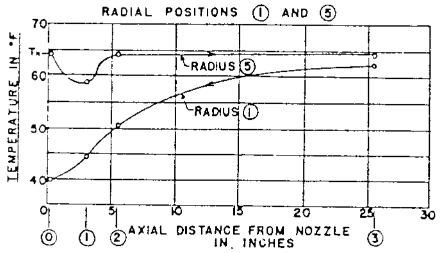

Fig. 4. Total temperature vs axial distance from nozzle at two radial position, $u = 0$.

The break in the curve of Figure 4 near traverse station 1 is believed to be due to radial mixing effects, since it has been shown that radial outflow occurs in this region (see Fig. 1, Cases B and C).

In the following analytical treatment radial mixing is neglected for simplification, and also because its effect is believed to be of a secondary nature. Figure 5 shows the simplified flow pattern assumed and also gives qualitative hypothetical temperature vs length curves to illustrate the temperature nomenclature used, and to clarify the theoretical analysis.

To satisfy the necessary overall heat balance in the absence of mechanical work or heat transfer to the environment

$$u(T_N - T_C) = (1 - u)(T_H - T_N) \qquad (1)$$

Or,

$$u(T_H - T_C) = T_H - T_N \qquad (2)$$

As in the case of conventional heat exchangers, a cooling effectiveness can be defined, which is the ratio of the heat transferred to the maximum possible heat transferable with the available temperatures, as follows:

$$N = (T_H - T_C)/(T_H - t_N) \qquad (3)$$

In Equation 3, t_N is the lowest available static temperature in the vicinity of the nozzle. If an ideal isentropic nozzle expansion is assumed, then

$$t_N = T_N - \Delta t \qquad (4)$$

Where

$$\Delta t = T_N/C\,(f(P))$$

Combining Equations 2, 3, and 4,

$$T_H - T_N = \frac{u\,\Delta t}{(1/N) - u} \qquad (5)$$

Then using Equation 1 with Equation 5,

$$T_N - T_C = \frac{(1 - u)\,\Delta t}{(1/N) - u} \qquad (6)$$

Equations 5 and 6 give the overall performance of a vortex tube as a function of the operating variables u and Δt and the cooling effectiveness, N. Knowing the performance data for a given vortex tube, N can be calculated from Equation 5 or 6. In the hope that N could serve as a correlating factor in vortex tube performance, the writer has calculated it for many of the operating points in the rather thorough data of R. Hilsch[1]. The results are given in Figures 6 and 7, where N is plotted vs u. Figure 6 gives four curves at constant nozzle pressures, P, varying from 2.5 to 11 atmospheres absolute for the No. 2 Hilsch tube which has the following principal dimensions:

Nozzle diameter	2.3 mm
Tube diameter	9.6 mm
Orifice diameter	4.2 mm

Figure 7 gives four curves all at $P = 11$ atm abs, but with four different orifice diameters, and one curve at

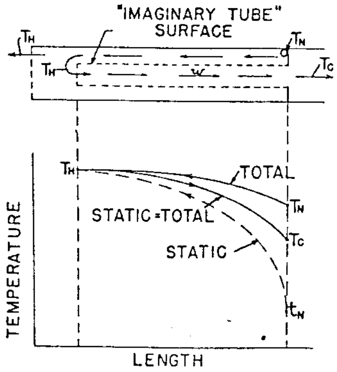

Fig. 5. Flow pattern and temperature nomenclature for theoretical derivation.

$P = 4$ atm abs with a 2.2 mm diameter orifice, all for the No. 1 Hilsch tube having the following principal dimensions:

Nozzle diameter	1.1 mm
Tube diameter	4.6 mm
Orifice diameter	varied from 1.4 to 2.6 mm

These data were calculated from points read off the published curves[1], which were reproduced to a very small size, and hence some accuracy has been lost in Figures 6 and 7. In Figure 6 the variations of N with u are well represented with straight lines. In Figure 7 the effect of orifice size on the variation of N is apparent. Comparison of the curves for the 1.4 and 1.8 mm orifices makes it appear that at some intermediate orifice size, say 1.6 mm, N would be virtually constant at a value of about 0.43.

Figure 8 is a plot of Hilsch's data (solid curves) for his No. 2 tube (same as used in Figure 6), giving temperature drop vs u for $P = 4$ and 11.

It now becomes necessary to point out a basic discrepancy which shows up in all the data of Hilsch when compared to the writer's experimental results. As Figure 8 indicates, the Hilsch data show the temperature drop as zero at $u = 0$. The writer's data indicate that the temperature drop is not zero at $u = 0$. Figure 4, for $u = 0$, shows that in the core just inside of the cold orifice, the air is in fact colder than it is for a value of $u = 0.20$, the latter value of u being according to Hilsch's data nearer the maximum temperature drop value. It is believed that this discrepancy lies in the location of and the method of the cold temperature measurement. Hilsch chose to measure the temperature of the metal wall of the tube on the discharge side of the orifice as an indication of cold air temperature. In Hilsch's words, "These measurements yield temperature differences which are certainly not too favorable, but will probably be too small." This must certainly be so, and particularly so as the cold flow approaches zero, at which point the metal temperature must be very near room air temperature. However, even if a temperature measurement is taken in or near the center of the cold tube on the discharge side of the orifice (as was done in the writer's tests), three side effects can increase the reading above that which exists in the center of the orifice, namely:

1) Heat pickup from the "cold" tube
2) Orifice edge leakage (see Fig. 1, Cases B and C)
3) Recirculation in the "cold" tube which can mix room air with the cold air (see Fig. 1, Case B)

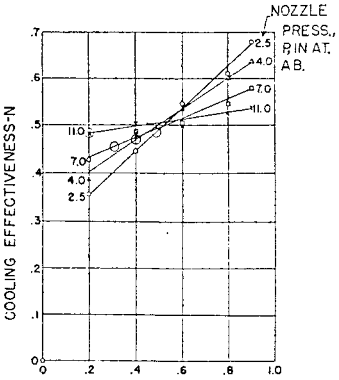

Fig. 6. Cooling effectiveness vs cold fraction for No. 2 Hilsch tube at four nozzle pressures.

Referring again to Figures 6 and 7, it can be seen that since Hilsch's data show zero temperature drop at zero cold flow, there is a point for each curve at $N = 0$ and $u = 0$, but in no case does this point fall reasonably on any of the curves. The writer believes that this apparent discontinuity in the cooling effectiveness, N, between $u = 0.2$ and $u = 0$ substantiates the aforementioned contention that a temperature drop exists at $u = 0$, even though it is reduced by the three side effects mentioned.

The larger circles shown on the curves of Figures 6 and 7 represent the points at which the temperature drop is a maximum according to Hilsch's data. This occurs in general between $u = 0.2$ and $u = 0.5$, depending on nozzle pressure and orifice diameter. It can be noted that the value of N at maximum temperature drop varies only from 0.45 to 0.48 for Figure 6, and from 0.41 to 0.46 for Figure 7. The higher average value of N for Figure 6 is attributable to a larger size tube (about twice the diameter) than that of Figure 7.

Now by assuming a constant value of N of 0.46 for No. 2 Hilsch tube (Figs. 6 and 8), Equation 6 becomes

$$T_N - T_C = \frac{(1 - u)\,\Delta t}{2.17 - u} \qquad (7)$$

Equation 7 represents in simplified form the performance of a vortex tube of the size and design of the Hilsch No. 2 tube. This equation has been plotted for $P = 4$ and 11 in Figure 8 (dashed curves). The difference between the solid and dashed curves lies in the assumption of constant cooling effectiveness, N, in Equation 7.

The next step is to look into a theoretical derivation of N, and to see if the values of N obtained from the test data could be reasonably attributed solely to a heat transfer process. In the following, an approximate derivation of N as a function of heat transfer coefficient, effective surface area, and flow is given.

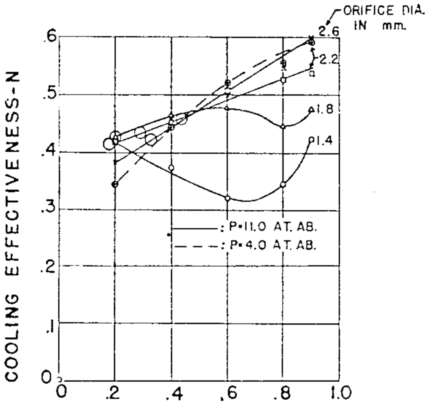

Fig. 7. Cooling effectiveness vs cold fraction for No. 1 Hilsch tube for four orifice diameters.

Referring again to the diagram of Figure 5, the rate of heat transfer from the core flow within the imaginary tube surface is

$$Q = wC\,(T_H - T_C) \qquad (8)$$

And also,

$$Q = AU\theta \qquad (9)$$

Where A is the effective surface area of the imaginary tube, θ is the mean static temperature difference, and U is the overall heat transfer coefficient. Combining Equations 8 and 9 with Equation 3 gives

$$\mathrm{N} = \frac{AU}{wC}\left(\frac{\theta}{T_H - t_N}\right) \qquad (10)$$

It is notable that due to the unusual nature of the outer helix flow in which static temperature increases more than total temperature due to the large negative velocity gradient, the conventional log-mean temperature difference is not applicable here for θ. However, by making the simplifying assumption that the temperature difference varies linearly with A, a very simple expression for N can be obtained which should be adequate since the effective area, A, is not known close enough to justify refinement in θ. Since the core flow is at a relatively low velocity, it is also assumed that the total and static temperatures are equal here. Then, referring to Figure 5,

$$\theta = (T_C - t_N)/2 \qquad (11)$$

Since $T_H - t_N = T_C - t_N + T_H - T_C$,

$T_H - t_N = T_C - t_N + (AU\theta)/(wC)$, using Equations 8 and 9.

And therefore, using Equation 11

$$T_H - t_N = 20 + \frac{AU}{wC}\theta \qquad (12)$$

Substituting Equation 12 in Equation 10 gives

$$N = \left(\frac{AU}{wC}\right) \Big/ \left(2 + \frac{AU}{wC}\right) \qquad (13)$$

Equation 13 is in the desired form. It is interesting to note the similarity between this equation and that for a conventional counterflow heat exchanger with equal heat capacity (wC) on both sides, for which

$$N = \left(\frac{AU}{wC}\right) \Big/ \left(1 + \frac{AU}{wC}\right) \qquad (14)$$

From Equation 13 it can be seen that for N to be 0.50, the approximate value calculated for the Hilsch tubes,

$$AU/wC = 2$$

If approximately representative values of A and w are used, the value of U required for this theory to match the actual performance is easily calculated. Taking for example, the No. 1 Hilsch tube (Fig. 7 data):

$A = 0.0113$ sq ft, based on an assumed effective length of 65 percent of the hot tube length and a diameter equal to the cold orifice (2.2 mm)

$C = 0.24$ Btu per lb, deg F

$w = 6.75$ lb per hr, which is the total nozzle flow at $P = 4.0$ atm abs.

Then

$$U = \frac{2 \times 6.75 \times 0.24}{0.0113} = 286 \text{ Btu per hr, sq ft, deg F}$$

As defined for the foregoing analysis, w is the flow inside the imaginary tube. Due to the radial outflow near the orifice which was shown to exist under most operating con-

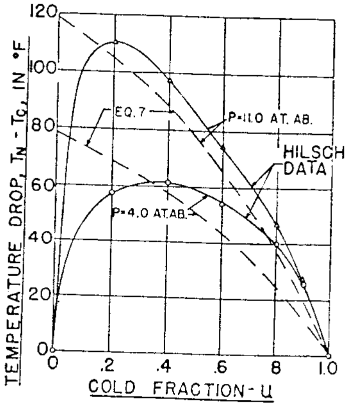

Fig. 8. Temperature drop vs cold fraction for No. 2 Hilsch tube at two nozzle pressures, by test and by theoretical equation.

ditions (see Fig. 1), the flow from the orifice is less than the value of w inside the tube. Thus even at $u = 0$ (no cold flow) w may be fairly large, but probably no larger than the nozzle flow. It is for this reason that the total nozzle flow is used in the foregoing evaluation of U. If the actual core flow, w, is less than the nozzle flow, then U required is proportionally lower. Also it can be seen that as P is raised or lowered, the U required will increase or decrease proportional to the flow, but this is compensated for by the available heat transfer coefficient varying almost proportional to the mass flow rate.

The mass flow rate in the preceding numerical example is about 46 lb per sec, sq ft in the core, which corresponds in a tube of 2.2-mm diameter to a film coefficient of about 140 Btu per hr, sq ft by a conventional heat transfer equation for flow in a straight tube[7]. This is one half of the required overall coefficient. The following two factors may be sufficient to raise the above heat transfer coefficient up to the value of U required by the heat transfer theory.

1) A large degree of turbulence due largely to the helical nature of the flows, which would raise the heat transfer coefficients. It is known, for instance, that for flow inside helically coiled pipe, heat transfer coefficients may be several times higher than in straight pipe for the same Reynolds number[7].

2) The absence of a solid wall at the interface between the core and the outer flow would seem to make it logical for the film coefficient to be based on the total relative axial velocity of the inner and outer streams. Since these flow countercurrently, their velocities are additive. Due to the absence of a solid tube wall there are not two film resistances, but only one resistance evaluated from the relative axial velocity. The effect of using a relative velocity in the preceding numerical example would be to raise the calculated film coefficient by about 30 percent, from 140 to 182.

Conclusions

The heat transfer theory of vortex tube action as presented in this paper is not complete in itself and must rely on empirical results in the performance equation (Eq. 6). The cooling effectiveness, N, has been shown to be a useful correlating factor. Although the required overall heat transfer coefficient is greater than can be accounted for by conventional calculations, the two are sufficiently within range as to be encouraging. The theory does not preclude the possibility that other mechanisms may be simultaneously acting to produce cooling of the air. Other theories have been published recently[8,9] which are of an entirely different nature, but which do not appear to have been based on prior tests concerning the nature of the internal flow processes. Discussion of these theories is beyond the scope of this paper.

The writer suggests a test setup in which, if any cold air were produced, the heat transfer theory would be greatly enhanced, as follows:

Run a vortex tube in which the "imaginary tube surface" of the preceding theory is replaced with a thin metallic tube which is rotated mechanically at an angular velocity comparable to that of the air at the tube radius.

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March 27, 1934. G. J. RANQUE 1,952,281

METHOD AND APPARATUS FOR OBTAINING FROM A FLUID UNDER PRESSURE
TWO CURRENTS OF FLUIDS AT DIFFERENT TEMPERATURES

Filed Dec. 6, 1932 3 Sheets-Sheet 1

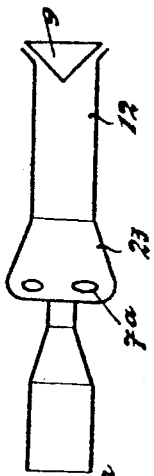

Fig: 9

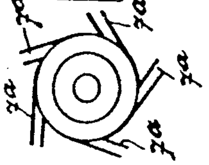

Fig: 10

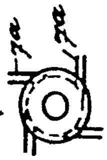

Fig: 12

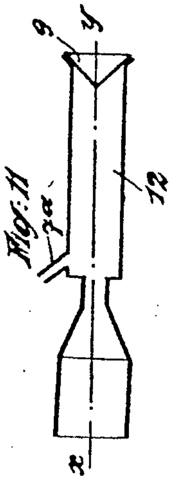

Fig: 11

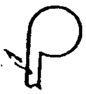

Fig: 2

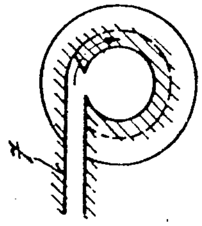

Fig: 6a

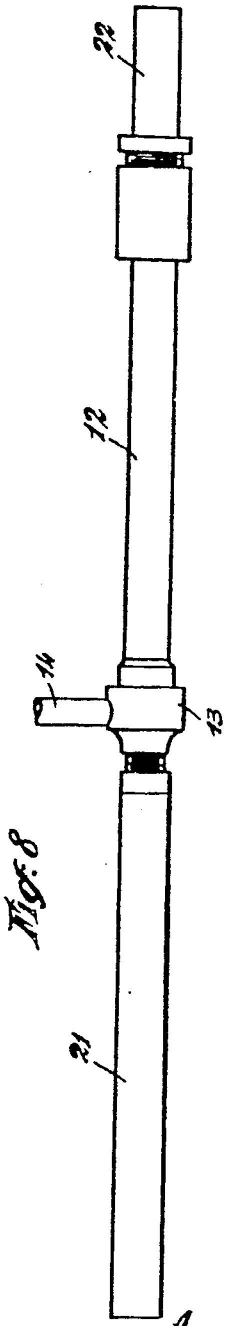

Fig: 8

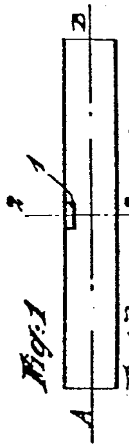

Fig: 1

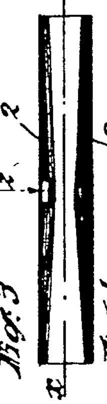

Fig: 3

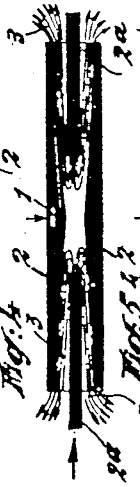

Fig: 4

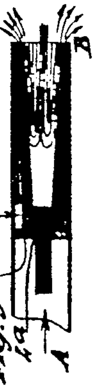

Fig: 5

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March 27, 1934. G. J. RANQUE **1,952,281**

METHOD AND APPARATUS FOR OBTAINING FROM A FLUID UNDER PRESSURE TWO CURRENTS OF FLUIDS AT DIFFERENT TEMPERATURES

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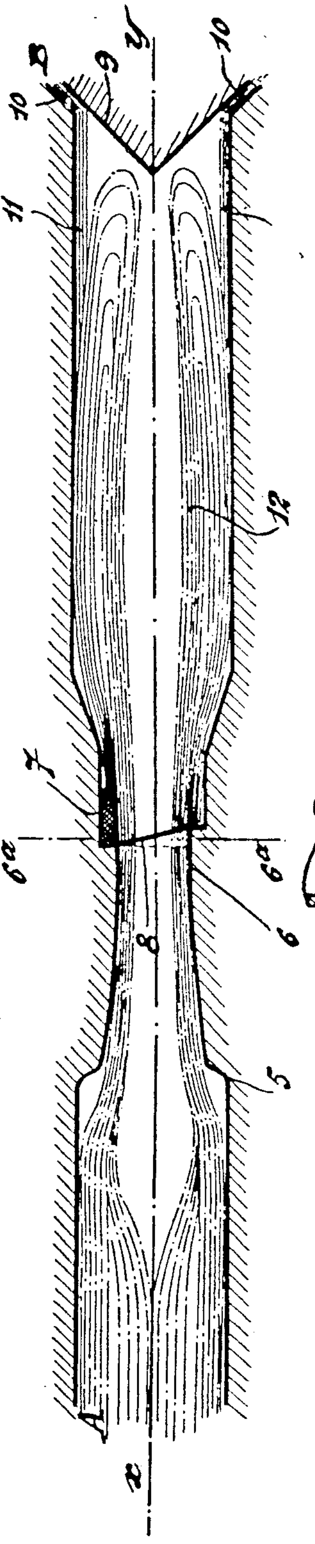

Fig. 6

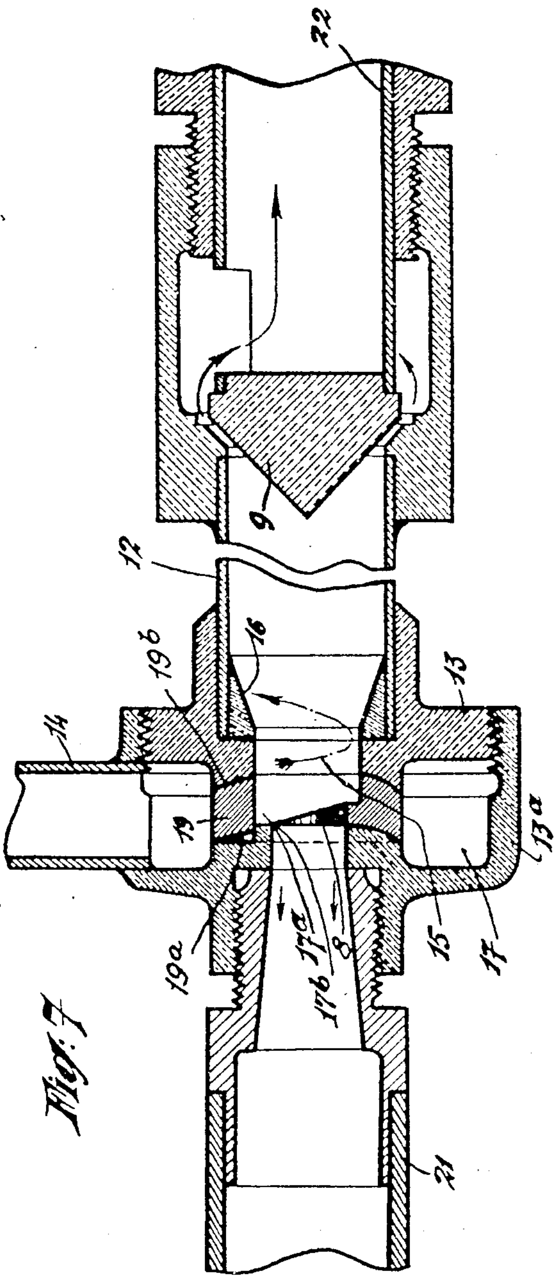

Fig. 7

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METHOD AND APPARATUS FOR OBTAINING FROM A FLUID UNDER PRESSURE
TWO CURRENTS OF FLUIDS AT DIFFERENT TEMPERATURES

Filed Dec. 6, 1932 3 Sheets-Sheet 3

Fig. 13

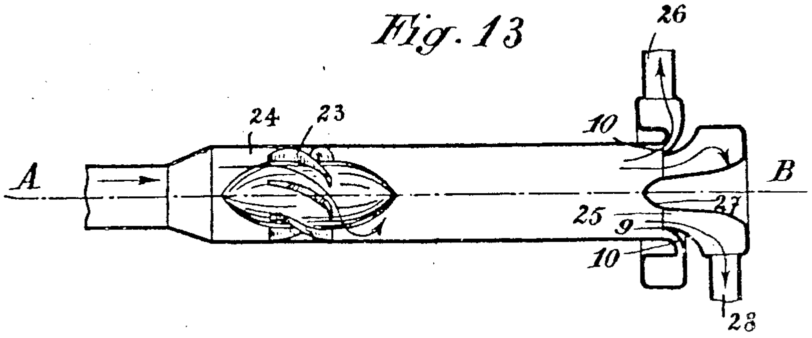
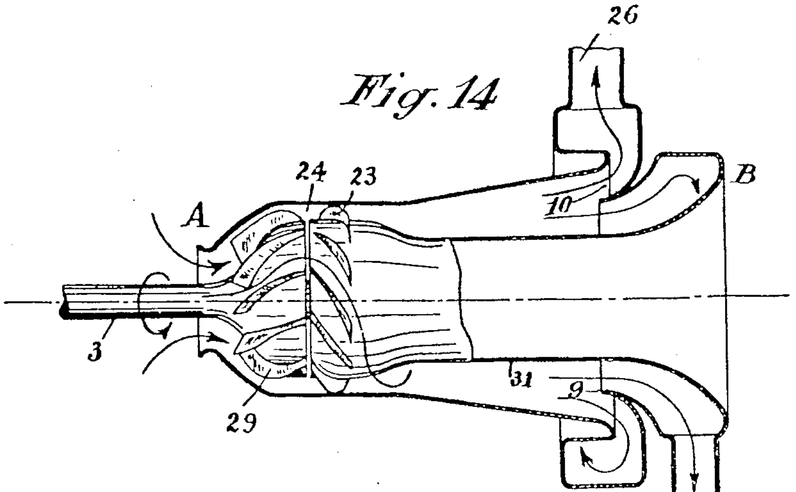


Fig. 14



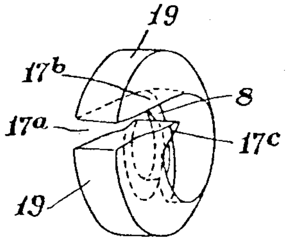


Fig. 7a

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Patented Mar. 27, 1934

1,952,281

UNITED STATES PATENT OFFICE

1,952,281

METHOD AND APPARATUS FOR OBTAINING FROM A FLUID UNDER PRESSURE TWO CURRENTS OF FLUIDS AT DIFFERENT TEMPERATURES

Georges Joseph Ranque, Montlucon, France, assignor to La Giration Des Fluides, Societe a Responsabilite Limitee, Montlucon, France, a company of France

Application December 6, 1932, Serial No. 646,020
In France December 12, 19

15 Claims. (Cl. 62—170)

The object of my invention is a method for automatically obtaining, from a compressible fluid (gas or vapour) under pressure, a current of hot fluid and a current of cold fluid, that transformation of the initial fluid into two currents of different temperatures taking place without the help of any movable mechanical organ, merely through the work of the molecules of fluid upon one another.

The method according to my invention consists essentially in dividing the fluid under pressure, which is admitted tangentially into a vessel having the shape of a body of revolution, into two coaxial sheets of fluid moving with a gyratory motion and reacting upon each other so as to produce, under the action of centrifugal force, the compression of the outer sheet by the inner sheet which expands, this compression absorbing a certain amount of work, which is evidenced by a rise in the temperature of the compressed sheet at the expense of the other sheet, which is thus cooled.

In a practical mode of carrying out this method, the fluid under pressure is introduced tangentially into a vessel having the shape of a body of revolution provided with axial orifices disposed on either side of the fluid inlet. Said fluid is suitably guided so as to give it a helical motion toward one of said orifices, the cross section of which is suitably restricted so as to produce a backward motion of a portion of the fluid toward the opposite orifice. This produces two sheets of fluids having opposite axial motions, the inner sheet expanding and compressing the outer sheet, thus supplying heat thereto. A current of hot fluid is thus received through the orifice of restricted cross section, while a current of cold fluid is received through the opposite orifice.

Another object of my invention is to provide an apparatus for carrying out the method above referred to. According to my invention, this apparatus comprises a chamber having the shape of a body of revolution the middle part of which is provided with one or more tangential inlet tubes for the fluid under pressure. Axial orifices are provided at either end of said chamber, one of said orifices, toward which the liquid, or fluid is directed through a suitable guiding with a gyratory motion, having a cross section smaller than that of the sheet of fluid, so that a portion of the latter is driven back toward the opposite orifice in such manner that it is caused to flow over the sheet of fluid that is applied against the wall of the chamber in question.

Preferred embodiments of my invention will be hereinafter described with reference to the accompanying drawings, given merely by way of example, and in which:

Figs. 1 to 5 inclusive are diagrammatical views illustrating the principle of my invention,

Fig. 2 being a sectional view on the line 2—2 of Fig. 1;

Fig. 6 is a diagrammatical view of an embodiment of my invention;

Fig. 6*a* is a sectional view on the line 6*a*—6*a* of Fig. 6;

Fig. 7 is a detailed view showing in axial section a practical embodiment of my invention;

Fig. 7*a* is a perspective view of the helicoidal guide;

Fig. 8 is a corresponding plan view on a smaller scale;

Fig. 9 is a diagrammatical elevational view of another embodiment of my invention;

Fig. 10 is an end view corresponding to Fig. 9;

Fig. 11 is a diagrammatical view of another embodiment of my invention;

Fig. 12 is an end view corresponding to Fig. 11;

Figs. 13 and 14 are diagrammatical views of two other embodiments of my invention.

The principle on which my invention is based is illustrated by diagrammatic Figures 1 to 5. Supposing, as shown in Figs. 1 and 2, that a tube A B is provided in its middle part with a tangential inlet pipe 1 through which a current of a compressible fluid (gas or vapour) under pressure is sent into said tube, said fluid is given in said tube a certain linear velocity parallel with the axis of said tube, said rectilinear movement being combined with a gyratory movement about the axis of the tube. The fluid flows toward both ends of the tube.

As the fluid is moving away from the inlet pipe, its rectilinear velocity, which is parallel with the axis $x\ y$ of the tube, increases, and its angular velocity decreases, so that the fluid spreads along the wall of the tube so as to form a sheet 2 having substantially the shape of a body of revolution about axis $x\ y$ (Fig. 3). In said sheet the molecules are subjected to a pressure which is the higher as they are at a greater distance from the axis of the tube, due to the action of the centrifugal force. At the same time, the flow of the fluid produces a substantial fall of pressure in the central zone of the tube, so that the outer air, which is at the atmospheric pressure, is drawn toward the central zone of the tube, thus forming two axial currents 2*a* (Fig. 4). When the outer air reaches said central zone, it is driven back toward the outside by the fluid moving with a gyratory movement, thus forming streams 3.

If an annular diaphragm 4, the free central opening 4*a* of which has a diameter equal to the minimum diameter of the zone in which a fall of pressure is produced, as shown in Fig. 5, is provided in the central part of the tube, on one side of the tangential inlet pipe, the fluid moving with a gyratory movement will flow only toward orifice B, carrying along with it the atmospheric air coming from both orifice A and arifice B.

The method and the apparatus according to my invention are based on the experimental facts that have just been stated.

In the embodiment shown in Figs. 6 and 6*a*, the apparatus consists of a chamber 5 having the shape of a body of revolution about axis x y, the middle part 6 of said chamber being of restricted cross section and being provided with a tangential inlet pipe 7 for the fluid (gas or vapour) under pressure. The inner wall of chamber 5 is provided, opposite the opening of said pipe, with a helical guiding surface 8. The orifice A of chamber 5 is freely opened, while the cross section of orifice B is restricted by a kind of frusto-conical diaphragm or deflector 9, so that the fluid under pressure, admitted through pipe 7, is only allowed to flow through an annular aperture 10, which is not sufficient for the amount of fluid fed thereto. The fluid under pressure admitted through pipe 7 and guided by helical surface 8 is simultaneously given a rectilinear motion which causes it to move within chamber 6 toward opening 10, and a rotary motion about axis x y. The sheet of fluid that is immediately adjacent the wall of the chamber flows out through said opening 10, while the remainder of the fluid, which is prevented from flowing out by diaphragm 9 is subjected to the fall of pressure existing in the central zone of the chamber and is given a backward motion toward orifice A. I thus obtain, according to my invention, a first sheet of fluid 11, moving with a gyratory motion along the inner wall of the chamber, from orifice 7 toward orifice B, and a second sheet of fluid 12 moving with a gyratory motion along the inner surface of the first mentioned sheet in an opposite axial direction, said second sheet of fluid consisting of the difference between the amount of fluid admitted through pipe 7 and the amount of fluid that is allowed to flow out through opening 10. Said sheet of fluid under pressure 12, which moves with a gyratory motion not along the rigid wall of chamber 5, but along the elastic surface of the first mentioned sheet of fluid, tends, on the one hand under the action of the centrifugal force, and on the other hand under the effect of the increase of velocity due to the expansion and to the rotation that take place, to compress the molecules of the first mentioned sheet of fluid. That compression absorbs a certain amount of work, which is evidenced by a loss of heat from the second mentioned sheet to the benefit of the first mentioned one. Consequently, the temperature of sheet 12 falls, while the temperature of sheet 11 rises. Finally, there is obtained through orifice 10 a current of hot fluid, and through orifice A a current of cold fluid.

The initial guiding of the fluid toward one of the orifices is necessary for practical purposes in order to obtain an accurate centering of the central zone of depression or fall of temperature. In the example above described, that guiding is effected through helical inclined surface 8. The following description will show that the same result could be obtained through other guiding means.

The adjustment of the cross section of the outlet orifice at B, which can be obtained through any suitable means makes it possible, by modifying the rates of flow at B and A, to vary the differences between the temperature of the initial fluid and those of the hot fluid and of the cold fluid escaping through outlet orifices B and A respectively.

If, for instance, the cross section of the orifice 10 through which the hot fluid is allowed to flow out is considerably restricted, the rate of flow of the hot fluid is diminished, but the rate of flow of the cold fluid is simultaneously increased so that the heat that is given out from one sheet to the other one causes a considerable rise of the temperature of the hot fluid but a small fall of the temperature of the cold fluid, as compared with that of the initial fluid.

Figs. 7 and 7*a* show a practical embodiment of my invention.

This embodiment comprises a cylindrical chamber 12 in which the interchange of heat takes place, and an annular distributing organ made of two pieces 13—13*a* which is provided with an inlet pipe 14 for the fluid under pressure. Said distributing organ comprises an inner cylindrical chamber 15 connected with cylinder 12 through a frusto-conical surface 16, and with annular conduit 17 of the distributing organ through a tangential passage 17*a*. The guiding helical surface 8 extends from one edge 17*b* to the other 17*c* of the orifice of said passage. The tangential passage 17*a* and the guiding inclined surface 8 are provided in a separate part 19, provided with conical surfaces 19*a*—19*b* for the centering thereof between parts 13 and 13*a* of the distributing organ. On the side opposite to cylinder 12 said distributing organ is connected with a cylinder 21 at the end of which the current of cold fluid is received, while the current of hot fluid passing through the annular orifice provided around conical diaphragm 9 is received through tube 22.

Figs. 9 to 13 show other embodiments of the means for guiding the fluid. In the embodiment of Figs. 9 and 10, said guiding is obtained through several tangential pipes 7*a* opening into a frusto-conical chamber 23 connected with the working chamber 12.

In the embodiment of Figs. 11 and 12, the guiding action is obtained through several pipes 7*a* opening tangentially into chamber 12, but which are inclined with respect to the axis x y of said chamber.

It should be well understood that it is not absolutely necessary, according to my invention, that the fluid under pressure should be admitted tangentially into a chamber having the shape of a body of revolution in which the fluid is divided into two coaxial sheets one of which receives from the other one mechanical work which is transformed into heat. What is necessary is to obtain an annular flow of the fluid moving with a gyratory movement and any means for obtaining that result may be obtained according to my invention. In particular, I may use to this effect directing blades disposed for instance in an inlet conduit coaxial with the chamber in which the interchange of heat takes place.

Furthermore, instead of being provided on either side of the inlet conduit, the axial orifices through which the two sheets of liquid escape may be disposed on the same side of said inlet conduit, the annular orifice for the outflow of the hot fluid surrounding the outlet orifice for the cold fluid. In such an arrangement, the two sheets have parallel axial movements in the same direction, which may be advantageous in some cases for reducing their mutual friction.

Such an arrangement is shown in Fig. 13 in which the fluid is admitted at one of the ends A of the chamber A B having the shape of a body of revolution and is given a gyratory movement by a plurality of blades 23 disposed in an annular tube 24. The other end B of chamber A B is provided with two concentric orifices 10 and 25 disposed in such manner that the outer orifice is limited by a diaphragm 9 so that the fluid moving with a gyratory motion from end A past blades 23 cannot escape entirely through said orifice 10. A part of said fluid is compelled to escape through the inner orifice 25, of smaller diameter, which corresponds to a zone of lesser pressure.

This causes an expansion of that portion of the fluid and it has been ascertained experimentally that said expansion starts as soon as the fluid leaves the directing blades and is continued as far as orifice 25. According to the laws of gyratory flow, said expanding sheet compresses the sheet that surrounds it and that flows out through annular orifice 10 and tube 26. In order to avoid parasitic entrainments, it is advantageous to give also to orifice 25 an annular shape by means of a deflector 27, along which the inner sheet flows before reaching tube 28. To sum up, tube 28 serves to the outflow of a portion of the fluid that is cooled by expansion with production of external work and tube 26 serves to the outflow of the remaining portion of the fluid, which is heated by compression.

Finally, instead of extracting the initial energy that is necessary for the working of the apparatus from a compressed air reservoir, it may be necessary in some cases to make use of mechanical energy for imparting a gyratory movement to the fluid and for giving it the superpressure that is necessary for its flow through the apparatus. To this effect, I may dispose, in concentric relation with the stationary blades that control the inlet of fluid, a plurality of movable blades which are mechanically actuated and are disposed in the same manner as the rotor of an air fan or of a compressor.

Such an arrangement is diagrammatically shown in Fig. 14 in which the initial energy of the fluid is not due to a preliminary compression in a separate apparatus but is imparted thereto in the apparatus itself by means of a rotor with blades 29 which is mechanically driven by a shaft 30.

In this embodiment all the other parts are disposed in the same manner as in the apparatus of Fig. 1, with the exception of deflector 27 which is replaced by an annular body 31 extending along the whole length of chamber A B, which is preferable when the diameter of the latter is relatively large.

While I have described what I deem to be preferred embodiments of my invention, it should be well understood that I do not wish to be limited thereto as there might be changes made in the arrangement, disposition and form of the parts without departing from the principle of my invention. It will be understood that it is advantageous to reduce the interchanges of heat between the various parts and between said parts and the outside by means of suitable heat insulating arrangements. Finally the adjustment of the difference of temperature between the hot sheet of fluid and the cold sheet may be obtained by modifying the ratio of the flows of the hot and cold fluids to the initial flow, which may be produced by modifying the sections or the inlet or outlet pressure of one of the three currents of fluid. In particular, in order to increase the temperature of the hot sheet, I may restrict the section left by diaphragm 9 for the outlet of said sheet or reduce the rate of flow by means of a valve disposed on the outlet pipe for the outflow of the heat fluid, or increase the initial pressure of the fluid admitted into the apparatus or again act on the section or the pressure at the outlet of the cold sheet.

What I claim is:

1. A method of obtaining from a current of a compressible fluid under pressure a current of hot fluid and a current of cold fluid which comprises causing said compressible fluid to flow with a gyratory helical motion along a surface of revolution, and dividing said fluid into two coaxial sheets moving along each other so that the outer sheet is compressed by the inner sheet and by the action of centrifugal force, whereby the work thus produced causes a rise in the temperature of the outer sheet and a fall in the temperature of the inner sheet.

2. An apparatus for obtaining from a current of a compressible fluid under pressure a current of hot fluid and a current of cold fluid, which comprises in combination, a chamber having the shape of a body of revolution, means for causing a current of the said compressible fluid under pressure to form a sheet of fluid moving with a gyratory motion along the inner wall of said chamber, and means for causing a current of fluid under pressure to form another sheet of fluid moving with a gyratory motion along the inner surface of the first mentioned sheet, with a relative movement with respect thereto.

3. An apparatus for obtaining, from a current of a compressible fluid under pressure, a current of hot fluid and a current of cold fluid, which comprises in combination, a chamber having the shape of a body of revolution provided with axial orifices at either end, means for introducing the said compressible fluid under pressure tangentially into the middle part of said chamber, means for helically guiding said fluid toward one of said orifices, and means for partly stopping the last mentioned orifice so as to leave only an annular passage for the outflow of the fluid, the other orifice being open to the atmosphere.

4. An apparatus according to claim 3 in which the last mentioned means are adjustable so as to make it possible to vary the cross section of the annular passage for the fluid.

5. An apparatus for obtaining from a current of a compressible fluid under pressure, a current of hot fluid and a current of cold fluid, which comprises in combination, a chamber having the shape of a body of revolution provided with axial orifices at either end, at least one tangential inlet pipe for the said compressible fluid under pressure opening into the middle part of said chamber, means for helically guiding said fluid from said pipe toward one of said orifices, and a deflector for partly stopping the last mentioned orifice so as to leave only an annular outlet passage for the fluid, the other orifice being open to the atmosphere.

6. An apparatus according to claim 4 in which the means for guiding the compressible fluid

consist of a member having a helically inclined surface located opposite the opening of said inlet pipe into said chamber.

7. An apparatus according to claim 4 in which the means for guiding the compressible fluid under pressure consist of a frusto-conical chamber coaxially connected with the first mentioned chamber, the apparatus comprising a plurality of fluid inlet pipes tangentially connected with said frusto-conical chamber.

8. An apparatus for obtaining from a current of a compressible fluid under pressure a current of hot fluid and a current of cold fluid, which comprises in combination, a chamber having the shape of a solid of revolution provided with an axial orifice at either end, a plurality of tangential inlet pipes for the said compressible fluid under pressure opening into said chamber, said pipes being inclined with respect to the axis of said chamber so as to guide the fluid toward one of said orifices, and a defector for partly stopping the last mentioned orifice so as to leave only an annular outlet passage for the fluid, the other orifice being open to the atmosphere.

9. An apparatus according to claim 4 in which said chamber has a restricted cross section between the opening of the inlet pipe and the last mentioned orifice.

10. An appartus according to claim 4 in which there is provided a distributing organ made of two parts located opposite said inlet pipe, the guiding means consisting of a ring provided with a tangential inlet conduit and with an inclined helical surface, which ring is inserted between said two parts of the distributing organ.

11. An apparatus for obtaining from a current of a compressible fluid under pressure a current of hot fluid and a current of cold fluid, which comprises in combination, a chamber having the shape of a surface of revolution, an inlet pipe for introducing a compressible fluid into said chamber, a plurality of directing blades located opposite said pipe for imparting to said fluid a gyratory motion along the inner wall of said chamber and means for dividing said fluid into two concentric sheets so that one of said sheets gives up a portion of its heat to the other sheet.

12. An apparatus for obtaining from a current of a compressible fluid under pressure a current of hot fluid and a current of cold fluid, which comprises in combination, a chamber having the shape of a surface of revolution, means for causing a current of the said compressible fluid under pressure to form a sheet of fluid moving with a gyratory motion along the inner wall of said chamber, and two annular orifices disposed at the same end of said chamber for dividing said fluid into two concentric sheets of fluid so that one of them gives up a portion of its heat to the other one.

13. An apparatus for obtaining from a current of a compressible fluid under pressure a current of hot fluid and a current of cold fluid, which comprises in combination, a chamber having the shape of a surface of revolution an inlet pipe for introducing the said compressible fluid under pressure into said chamber disposed coaxially with said chamber at one end thereof, a plurality of directing blades in said chamber disposed opposite said inlet pipe for imparting to said fluid a gyratory motion along the inner wall of said chamber, and two annular orifices disposed at the opposite end of said chamber coaxially therewith for dividing said fluid into two concentric sheets one of which gives up a portion of its heat to the other one.

14. An apparatus for obtaining from a current of a compressible fluid a current of hot fluid and a current of cold fluid, which comprises in combination, a chamber having the shape of a surface of revolution, mechanical means for driving said compressible fluid into said chamber, directing means for imparting to said fluid a gyratory motion along the inner wall of said chamber, and means for dividing said fluid into two concentric sheets so that one of them gives up a portion of its heat to the other sheet.

15. An apparatus according to claim 14 in which the mechanical means consist of a fan disposed opposite the inlet end of said chamber.

GEORGES JOSEPH RANQUE.

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BIBLIOGRAPHY of the VORTEX TUBE

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When Georges Joseph Ranque obtained his patent on the vortex tube in 1934, he raised questions that still remain unanswered today. Recent interest in the tube has shown a need for a bibliography on this subject. There is still much repetition of work due to ignorance of the available references. This bibliography is not represented as being complete, however its scope is stated in the paragraphs preceding each section.

The bibliography is divided into five sections: American Periodicals, Theses and Term Papers, Government Reports and Patents, Industrial Reports, and Foreign Periodicals.

The periodicals in the first section are available in most of the larger public libraries with the exception of P-1 and P-4 which can be obtained from their publishers. The remaining publications may be obtained from the publishers or back number magazine outlets if a personal copy is desired.

The theses and term papers can probably be obtained by the interlibrary exchange method.

The government reports and patents may be obtained from their respective departments.

The industrial report, C-1, is controlled by the General Electric Company.

The foreign periodicals of the final section should be available at most of the larger public libraries. Most of the articles are in the native language of the authors.

American Periodicals

Since its introduction into the United States, the vortex tube has received publicity in many American technical periodicals. Most of these references were found by the use of Indexes.

Short descriptions of the tube may be found in P-3, P-5, P-6, P-10, P-14, P-15, P-16, P-20, P-21, and P-24.

Discussions of the theory of operation may be found in P-1, P-4, P-5, P-7, P-8, P-11, P-12, and P-19.

Mathematical analyses are included in P-1, P-4, P-5, P-7, P-17, and P-23.

Reports of experimental work are in P-8, P-13, and P-23.

Applications of the vortex tube may be found in P-2, P-8, and P-22.

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